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IRON AND STEEL ENGINEER

NOVEMBER • 1941

VOL. XVIII NO. XI

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Published Monthly by

ASSOCIATION OF IRON AND STEEL ENGINEERS

Entered as second-class matter January 25th, 1924, at Pittsburgh, Pa., under the Act of March 3rd, 1879

Statements and opinions given in articles and papers appearing in "Iron and Steel Engineer" are the expressions of contributors, for which the Association of Iron and Steel Engineers assumes no responsibility.

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Single Copy Foreign Prepaid $2.00 Single Copy $1.50

Subscription price in U. S. & Canada ..... $ 7.50 per year

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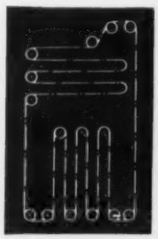

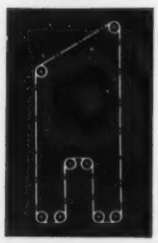

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THE HOT COIL CONVEYOR

By J. E. McBride, Vice President

PALMER-BEE COMPANY

DETROIT, MICHIGAN

Presented before A. I. S. E. Detroit Section, November 12, 1940

▲ MASS PRODUCTION, as we know it today, had its origin in the automobile industry. Other industries followed suit and, to supply these industries with their enormous requirements, steel mills had to resort to mass production for strip.

The advent and development of the continuous wide strip mill, with its great production capacity, brought with it a major material handling problem.

Without a doubt the most spectacular and interesting sight in any industry today is to witness the transition of a hot slab into strip; to watch its passage through about 1000 ft. of travel, from heating furnace to scale breaker, to roughing stands, to finishing train, and to emerge from the last finishing stand a wide glowing hot ribbon of steel, moving swiftly over the long runout table, to disappear with a swish into the coiler at a possible speed of 2000 ft. per min.

Within the space of three minutes a six-ton slab has become a beautifully formed, tightly rolled coil of strip. It is then pushed from the mandrel on which it has been wound onto the subject of our discussion—the hot coil conveyor.

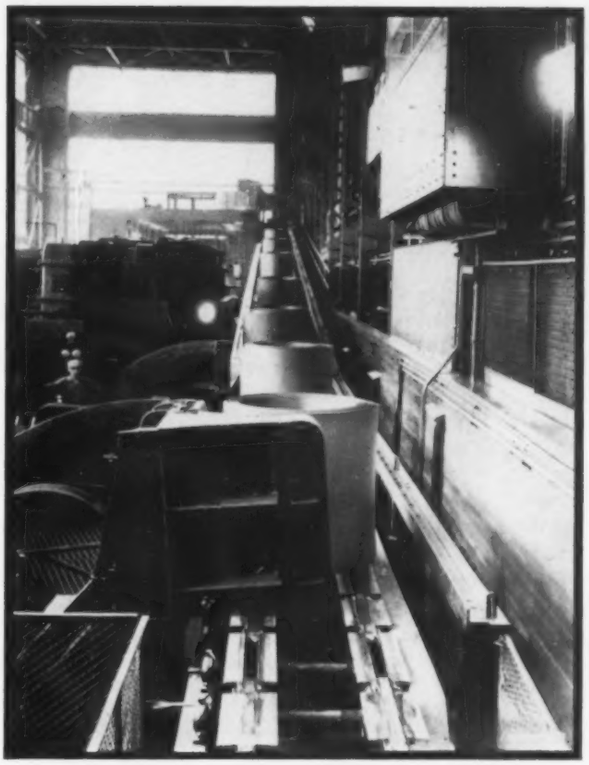


Figure 2 (above)—A hot coil conveyor consisting of long pitched chains supported by flanged drums operating in roller bearings.

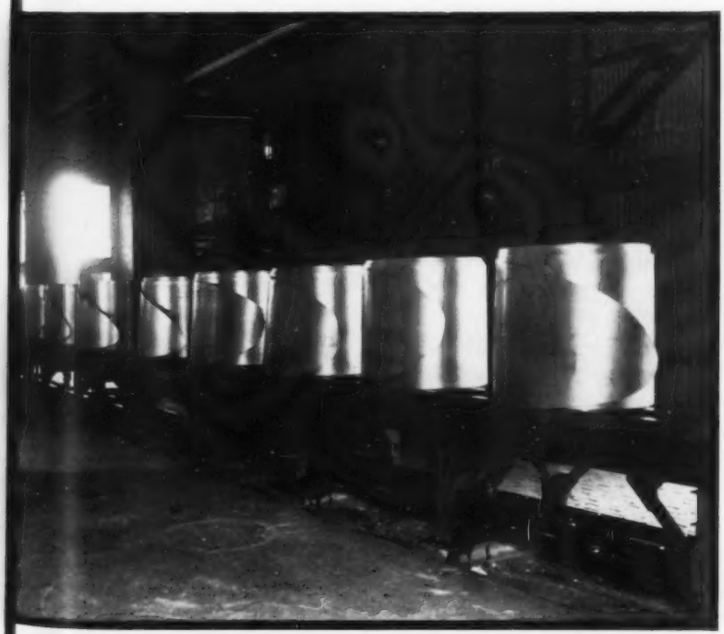


Figure 1 (left)—The sliding chain type of hot coil conveyor is simple and rugged, but requires heavy drive mechanism.

The use of conveyor equipment has become so diversified, and the word conveyor itself so broad in meaning, that I have decided to confine my remarks to one steel mill material handling problem only—the hot coil conveyor.

It is the most interesting and colorful of all material handling in the modern strip mill. Its place is of utmost importance in the general scheme, for it is the vital life line between the hot mill and coil storage.

We will therefore leave all other material handling subjects, such as sheet transfers, slab transfers, pilers, continuous pickling, feeder tables, sheet processing, live roller chain tables, etc., to some future time and to

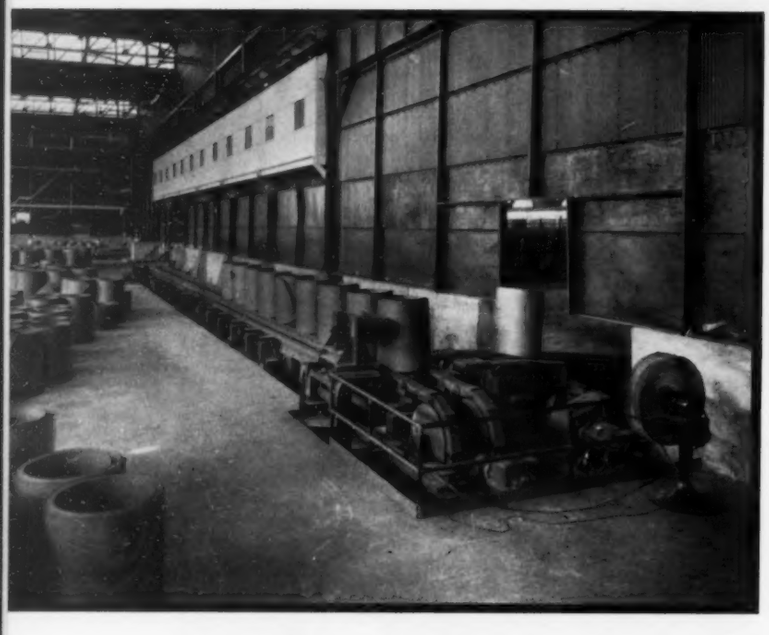


Figure 3—Right-angle transfer with "all-in-one" hot coil conveyor.

someone qualified to present their many interesting features.

The mechanical handling of hot coils begins at the downender or tilter, which receives the coils as they are stripped from the mandrel of the coiler at the end of the runout table, and ends where the coils are stored prior to shipment, or uncoiled for processing. This may be a distance of a few hundred feet or 1000 ft. or more.

Figure 4—View showing down-ender from coilers to "all-in-one" hot coil conveyor.

The conveyor receives the coil a considerable distance below the mill room floor, due to the step down from runout table to coiler, from coiler to down tilter and from down tilter to conveyor.

In the layout the receiving conveyor may run parallel to the mill and receive its load over the side, or it may run at right angles to the mill and receive its load in line with the movement of the downender. This latter arrangement is the most common and the loading is simpler to perform mechanically.

This receiving conveyor which runs on about a seven degree incline up to the mill room floor must transfer the coils through a 90 degree turn to a level line of suitable length for transportation to storage.

This level line may be long or short, it may run either direction from the transfer point or be reversible. It may be of one construction or a combination of several types of conveyor.

From level conveyor to storage the transfer is usually accomplished by a magnet crane which can handle two average size coils at once or the coils may be discharged over the head end to a downender for ram truck distribution.

While the hot coil conveyor is used chiefly for transportation, it has the added advantage in that the coils may be measured for width and gauge while on the conveyor, and cooled while in transit for better and safer handling.

The design of the hot coil conveyor must provide for automatic loading at the receiving point, which is at the downender from the coiler. It must provide automatic 90 degree or 180 degree transfer from one conveyor to another. It must provide for automatic weighing and recording the weight of the coils while on the conveyor. It must guard against damage to coils while in transit or smearing them with grease. It should provide for a minimum of manual operations, if for no other reason than on account of the intense heat. It should be free from any chance or liability of break down for as we have pointed out it is the vital life line between the hot mill and coil storage. It should have an excess factor of safety to withstand shock due to loading at the receiving end, and the impact of the magnet at the unloading end. It must provide for a capacity of from 100 to 120 coils per hour.

These specifications for design are very important. Today the hot coil conveyor must handle coils of maximum width, about 100 in.; maximum weight, about 15 tons; maximum heat, about 1200 degrees F.; maximum load, about 5000 lb. per lineal foot of conveyor. It may be of interest to know that the hot coil conveyor at its maximum capacity carries a load per foot three times more than the average fully loaded freight train per foot.

Like any other machine the hot coil conveyor has been developed through a number of stages in which trial and error, and survival of the fittest has played a part. In this development advantage has been taken of alloy steels, heat treatment, precision bearings, etc., to provide for strength, durability, smooth running, length of travel, resistance to shock and saving of power.

There are six different and distinct designs of hot coil conveyors in use in the fourteen or more modern continuous strip mills in this country today. In five of

these designs the fundamental principle is the same, in that the conveyor consists of two parallel chains, spaced on centers 24 in. to 30 in. apart, and not connected in any way. There is a clear space between the inside edges of the two chains of approximately 18 in. to permit the arm of the downender from the coiler to deposit the coils evenly onto the two chains. The top of each chain should present a wide smooth surface on which the coils rest. Here the similarity of the five designs ceases, and I will describe briefly the six different designs in the order in which they have been produced and installed in the various mills.

First is the sliding chain design, where two heavy block type chains slide in channel shaped tracks, fitted with wearing strips and mounted on eye beam stringers, the coils resting directly on the chain links. While this was the earliest design, it is used in some of the most recent installations. It has the following advantages: simple in design, consists of few parts, withstands shocks, and is easily and quickly repaired.

On the other hand, the length of conveyor handled by one drive and motor is limited to about 100 ft. for medium weight coils, and to about 50 ft. for maximum weight coils. These limitations are due to sliding friction, where for a lubricated track we use 20 per cent of the total moving load while running and 50 to 100 per cent for starting from a position of rest, to determine the chain pull. These percentages are used where the load pressure on the chain is about 200 lb. per sq. in. If the pressure is increased, the coefficient of friction is increased.

While the sliding chain is simple and rugged it is quite evident that due to the sliding friction a large part of the saving in cost of conveyor medium over the more modern designs must be put back in an extra heavy drive mechanism and motor, and through a long distance the more frequent use of drives and transfers still further adds to the total cost.

This type of conveyor must slide on a film of grease which to some extent finds its way on to the ends of the coils but whether this is detrimental I am not in position to say. It is in use at Bethlehem Steel Company, Jones & Laughlin Steel Corporation, Great Lakes Steel Corporation, Wheeling Steel Corporation, Granite City Steel Company, and others.

The first attempt to design away from the sliding chain to obtain the mechanical advantage of the wheel and axle and anti-friction bearings, resulted in placing a bed of commercial ball bearing rollers under each strand of what had been the sliding chain. This has been called the gravity roller design.

However, several objections to this construction were soon noticeable. The cost of power lubrication, eight points to every foot of conveyor, was too great. The construction was too weak to withstand shock and the impact of the magnet crane. It had too many parts.

Installations of this type are to be found in the mills at Youngstown and Gary.

The next step in advance was to take the original type of sliding chain, lengthen the pitch from 12 in. to 24 in., provide a wider top surface to support the weight of the coils, carry the chains on large diameter flanged drums mounted on through shafts with ends supported in self-aligning roller bearings and spaced 22 in. apart or a little less than the chain pitch.

This design met all the demands mentioned earlier in this paper. There was economy in the long pitch. There was about one lubricating point to each foot of conveyor. It has been built over 500 ft. in length with one drive. This type can be seen in operation at the Ford Motor Company, and, for the later discussion, we will call this the Ford type.

As in the design of other mechanical devices of importance, those of us concerned kept putting forth still further efforts for improvement to the hot coil conveyor.

The next idea was to further simplify the construction and combine all rollers and anti-friction bearings into the design of the chain itself. This was successfully done. The economical pitch of chain was set at 18 in. The width of top supporting plates at about 6 in. The rollers at the articulation joints of the chain were made large in diameter and wide in face (approximately 7 in. x 4 in.) and fitted with precision roller bearings.

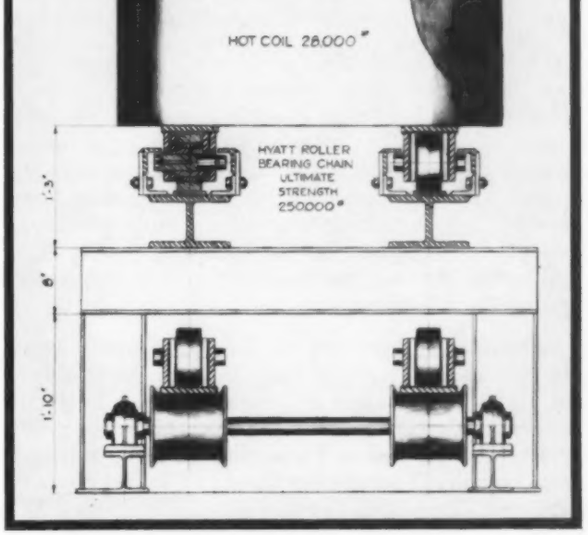


Figure 5—In the "all-in-one" conveyor, all rollers and anti-friction bearings are combined into the chain itself.

This made a compact design with only the normal number of parts required in any roller chain; but made with alloy heat treated steels, hardened and ground, and increased in size to withstand maximum loads and shocks. Lubrication was simplified in that it was infrequently required and the lubrication points traveled past either an automatic or a manually operated lubricator. For identification, we will call this type the "all-in-one."

In this and the following design we use a coefficient of 2 per cent of the total moving load to determine the chain pull, an advantage of 10 to 1 over sliding friction.

Republic Steel Corporation at Cleveland, the Irvin mill of Carnegie-Illinois Steel Corporation, and one of the Bethlehem plants use this construction.

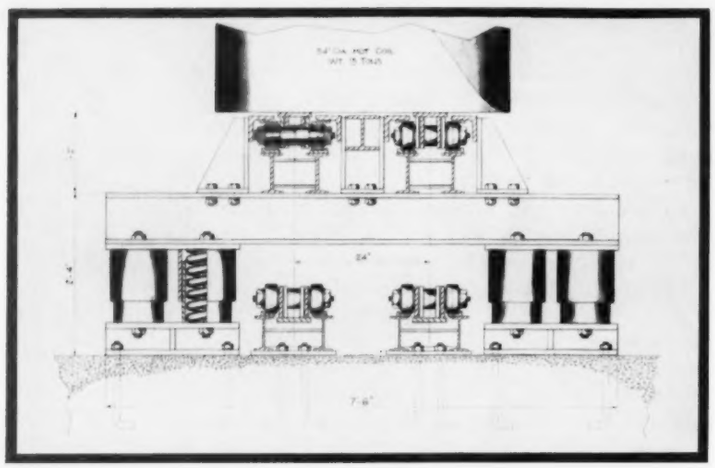
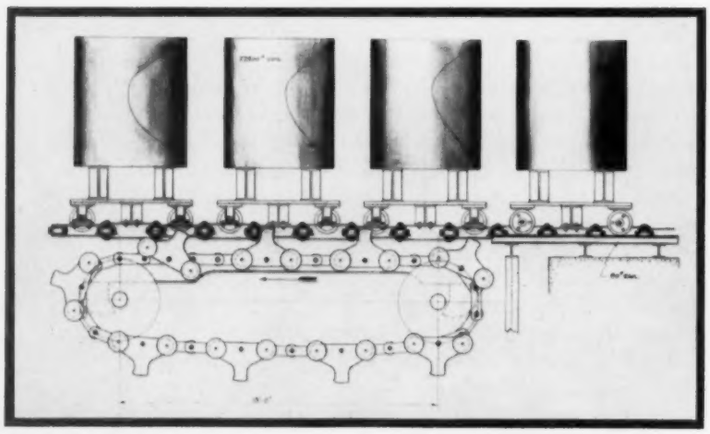


Figure 6—The "all-in-one" conveyor can be modified by using two outboard rollers rather than one in the center of the chain.

A modification of the "all-in-one" conveyor is to place the precision roller bearing rollers on the ends of extended chain joint pins, making two outboard rollers in place of one in the center of, and an integral part of the chain. This construction has some advantages and has met with considerable favor. The rollers are more accessible; can be quickly removed and replaced without dismantling the chain; has the appearance, at least, of greater stability; and can be easily guided on its track through the unloading area.

As compared, however, to the "all-in-one" design, this type has more parts, twice the number of rollers, four times the number of bearings, three times the number of lubrication points. We call this the outboard roller type. It is used at Tennessee Coal, Iron and Railroad Company, Youngstown Sheet and Tube Company, and Great Lakes Steel Corporation.

Figure 7—This "carry-all" type of conveyor, with caterpillar drive, operates in the shape of a long narrow loop.



In presenting the sixth and last design of hot coil conveyors, I wish to explain that it is basically different from the other five just described.

It was designed to meet a condition where the two parallel chain type could not be used. It had, of course, to meet all the requirements for performance previously mentioned. In this case the conveyor operates in a horizontal plane in the shape of a long narrow loop about 800 ft. of endless conveyor driven with two 50 hp. caterpillar drives.

The object is to cool the coils for a period of an hour through over 600 ft. of travel before they are taken to the upper level and piled on their sides. We call this the "carryall" type. It consists of a heavy single chain of 24 in. pitch, ultimate strength 250,000 lb. The chain is set on edge with the joint pins vertical to permit flexing of the chain and conveyor in a horizontal plane. Four-wheel cars or trucks spaced 6 ft. apart are attached to the chain links, and are so designed that the truck centers are always radial with the center of the curve

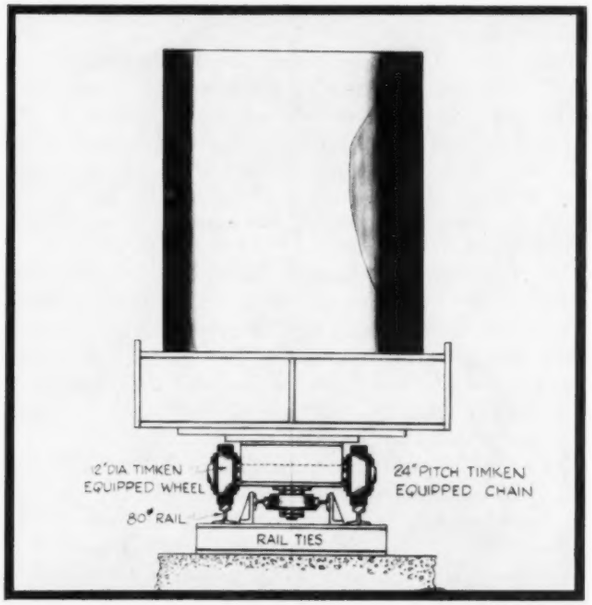


Figure 8—Cross section through "carry-all" coil conveyor designed for use where two parallel chains cannot be used.

when making a turn at the ends of the loop. This construction permits making the turn with a short radius and prevents chatter of the truck wheels on the track when making the turn.

The conveyor is loaded by two retractable down tilters at the coilers and unloaded by a retractable tilter at the end of the cooling period. The conveyor will run a distance of 6 ft., load and unload simultaneously and repeat. These operations are automatic by electric control.

The unloading tilter delivers the coils on their sides over a weigh scale to an inclined cradle type conveyor which runs to the mill room floor and to storage where

the coils are removed by a crane and piled on their sides.

This "carryall" type will be in operation early next year at the Weirton Steel Company.

It has been pointed out that in most installations the coils are loaded on to an inclined conveyor running at right angles to the mill and up to the mill floor where they are delivered through an angle of 90 degrees to a long cooling conveyor running parallel to the mill and to storage.

This 90 degree turn is an important part of the system. It must deliver coils from one conveyor to the other without injury, without shock and without disturbing their stability, especially where the wide coils are concerned. There are two methods of making this transfer. First, by using two short caterpillar flat top chains between the main conveyor chains and flush with their tops.

These pick up the coils a few feet back of the head end of the inclined conveyor and extend a few feet beyond in order to bridge the space where there is little support. These short caterpillar chains keep the coils moving steadily on to a few gravity rolls mounted on a pivot arm. These rolls drop through and deposit the coils onto the storage conveyor chains.

Second, the 90 degree and 180 degree transfer is made with power driven tapered rolls, presenting a level top surface. Gear head motors are used and every other roll only is driven.

A study of the two methods indicates there is not much to choose between the two. Both accomplish the same purpose equally well and overcome a rather serious difficulty at this point.

As a further aid to withstand shocks of the unloading magnet, it has been found advisable to mount the conveyor substructure at the unloading points on springs. These springs are of heavy design, slightly pre-compressed and permit a little give at the impact of the magnet, thus protecting the equipment from gradual damage.

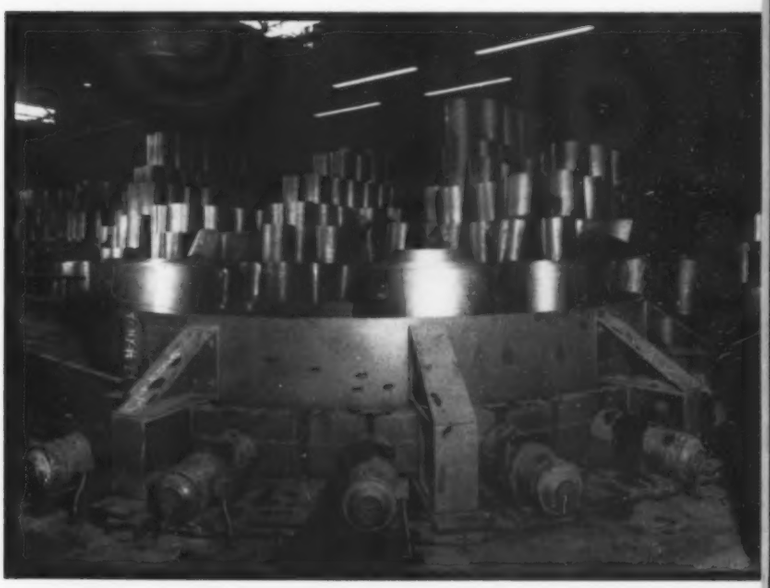
I have tried to bring before you the layout, design and development of what is recognized as a very important link in the continuous chain of events in the modern strip mill—a chain of events that begins at the slab heating furnace and terminates at the processing department or the storage for coils.

The answer to this major material handling problem as far as it has been reached today is the result of the combined efforts of the steel mill engineers and the conveyor manufacturers' engineers. In my opinion the best solution could not have been reached by either one without the help and coöperation of the other. One with years of experience in connection with his own mill and with his operating department, knows what to accept and what to reject and to offer many practical and constructive ideas. The other with his years of experience in designing, selling and acquiring knowledge in contact with many steel mills and their engineers, brings to his clientele the best available design and what is new in the industry. There is no doubt that the product of many minds with varied experience is of the greatest value in handling a difficult subject.

Figure 9 (left)—This retractable down-ender from strip coiler to conveyor operates through universal couplings from motor and reducer.

▲

Figure 10 (below)—Two parallel hot coil conveyors may be connected by a 180 degree motor driven tapered roller transfer.



Some Features of Steel Plant POWER

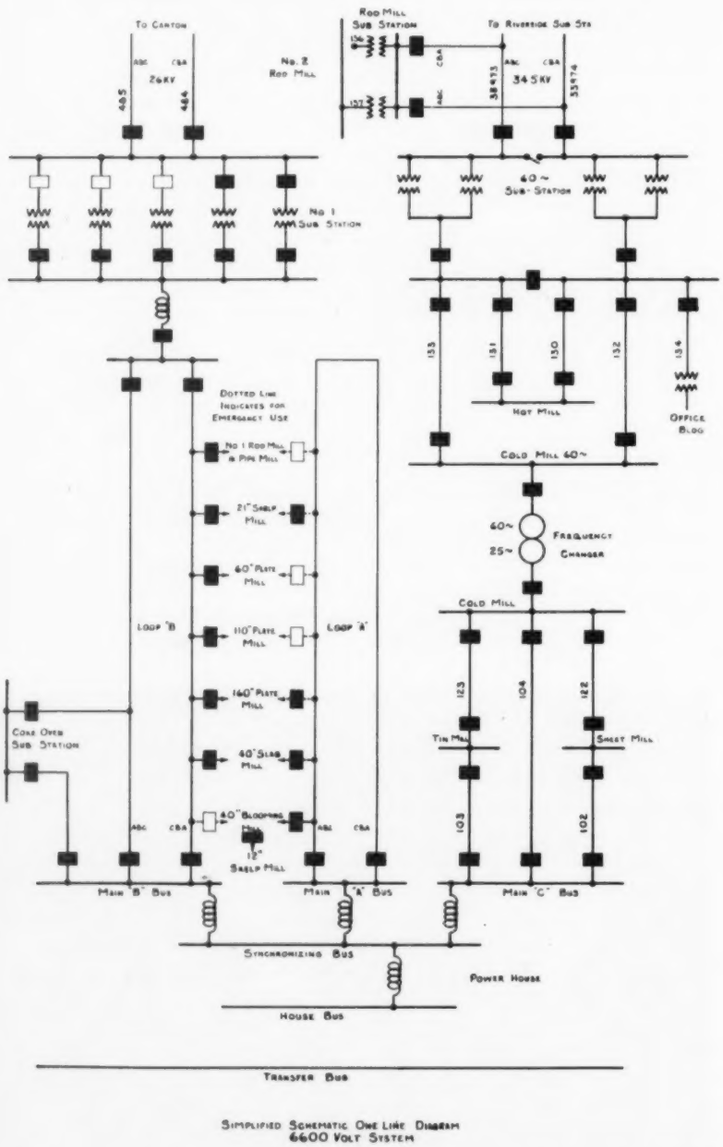
By F. O. Schnure, Supt. Electrical Department

BETHLEHEM STEEL COMPANY

SPARROWS POINT, MARYLAND

Presented before A.I.S.E.

Philadelphia Section, April 5, 1941



Simplified Schematic One Line Diagram
6600 Volt System

Figure 1—Diagram showing scheme used in installing 20,000 kw. frequency changer in plant power system.

▲ THE UTOPIA of every power operator is a system without any outages. While this, like the apparent convergence of railroad rails in the distance, is impossible; nevertheless, by the development of faster relaying and operation of circuit breakers, by the thoughtful application of lightning arresters and capacitors, by observing the effects of lightning strokes on transmission lines of varied construction, and countless other phases, engineers have improved the chances for uninterrupted service.

A few of the schemes recently developed have been applied in this plant and it is herewith the purpose to mention them, whether or not their value has been definitely demonstrated.

In order to coördinate the operation of a 20,000 kw. frequency changer, with the steel plant power system and with that of the power system serving the plant, it was necessary to install multi-conductor control cable between the frequency changer and the power house, some 8000 ft. away. With comparatively small expense it was possible to add the few additional conductors required to operate the pilot wire differential relays applied to nine feeders which connect as a unit, the power house, the frequency changer and a cluster of mills. One day last summer a locomotive crane swung into one of the circuits and the relaying cleared the fault so fast that only a few strands of the conductor burned and with no interruption to service. While it is difficult to evaluate what might have happened, the ability to restore that feeder to immediate service went a long way towards paying the initial cost.

The improvements in protective relaying would be of little benefit if the speed of circuit breakers themselves had not been improved. Along with that feature goes the responsibility of keeping the interrupting duty on a growing system within the capacities of the existing

Figure 2—Transmission lines may be protected from lightning strokes by two ground wires placed above the circuits.

TRANSMISSION PROTECTION

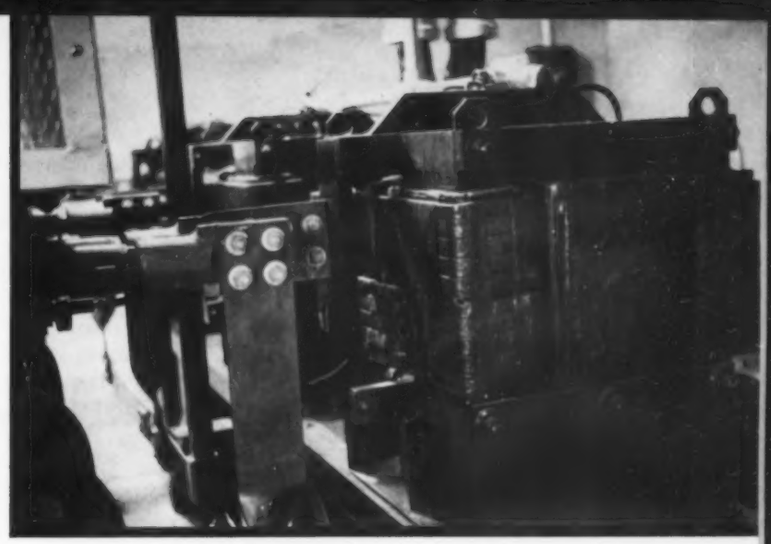


Figure 4—Saturable core reactors may be used on the secondary of lighting transformers to compensate for voltage variation.

equipment. The addition of the frequency changer in this case required further sectionalizing of the power house bus, following the general scheme shown in Figure 1, in which the "C" bus represents the change. It is the aim to keep the interrupting duty of the entire system within 500,000 kva.

Where power is used by large units the application of the proper capacity circuit breaker is not a serious consideration. However, where the units are comparatively small and yet too large to be economically served from a low voltage transformer bank it is more economical, considering first cost and power loss, to install current limiting reactors and use circuit breakers of lower interrupting capacity, than to install high capacity breakers. This practice has another advantage in that it allows the use of circuit breakers that have been removed from certain locations where system growth rendered inadequate their interrupting capacity.

The 25 cycle portion of this system consisting of 50,000 kw. generation and 37,500 kva. in purchased power, is operated with the wye of the purchased power transformers solidly grounded. Although some thought has been given to the installation of current limiting reactors to limit the ground current and subsequent transient voltage disturbances, nothing has been done due to the difficulty in evaluating the possible benefits.

However, on the 60 cycle portion of the system served by 60,000 kva. purchased power transformers, a current limiting reactor with shunting resistor was installed to limit the fault current to 2000 amperes. The shunting resistor's function is to eliminate certain transients that may arise from the set-up. To date, this installation has not yet been called upon to function.

Figure 3—Ground wires above circuits are placed high enough to cover them with an angle not over 15–20 degrees from vertical.

"The art of protecting transmission lines against lightning has advanced considerably in the last decade, but there are some phases of the problem which are still obscure." The above thought was expressed by an author of a paper presented at the recent A.I.E.E. convention in Philadelphia.

The transmission problems found in a steel plant, because of the short distances involved and the comparatively low voltages, are not the same as experienced on the present day high voltage transmission lines that ramble over the country. Nevertheless, repeated lightning strokes on some of our exposed 6600 volt lines have led us to try some of the ideas used on the big systems. One was the placing of two ground wires above the circuits high enough so that they would cover the circuits with an angle of not more than 15 to 20 degrees from the vertical (Figure 2). A unit of this type of installation on the 25 cycle system has gone through one season without a stroke where strokes previously had been frequent. Another unit on the 60 cycle system has not been struck since it was installed three years ago (Figure 3).

Another section of line virtually in the shadow of blast furnace stacks has been shielded by the installation of rods, consisting of 2 in. galvanized iron pipe, extending from the tower 10 to 15 ft. above the conductors and solidly grounded. While it takes several years to accumulate comparative operative data, experience to date indicates an improvement.

Along with the foregoing experiments has been a strong effort to maintain a low footing resistance on all towers by either driving ground rods or burying cable at a suitable depth where moisture was apt to be a permanent condition.

Voltage variation on lighting circuits is apt to be encountered where large fluctuating demands for power, such as strip mills, must be served. This variation may run as high as six volts and is easily noticeable. To correct this situation in the General Office Building, a saturable core reactor was installed on the secondary of three 150 kva., 6600/120 volt, 60 cycle transformers

(Figure 4). Each single phase regulating transformer consists of two transformers. The primaries are in series and connected across the supply line. The secondaries are in series opposition and in series with the load. One unit acts to increase the load voltage, the other to decrease it. Regulation of the load voltage is accomplished by changing the relative saturation of one core with respect to the other by circulating different values of direct current in the auxiliary d-c. coils on the center core legs. Whichever unit is the more saturated is the lesser effective; and the lesser saturated, the more effective.

The results shown on the curves in Figure 5 are satisfactory.

DISCUSSION

PRESENTED BY

L. V. BLACK, *Superintendent,* Electrical Department, Bethlehem Steel Company, Bethlehem, Pennsylvania

G. O. VAN ARTSDALEN, *Superintendent of Maintenance,* Henry Disston and Sons, Inc., Philadelphia, Pennsylvania

J. C. REED, *Electrical Engineer,* Bethlehem Steel Company, Steelton, Pennsylvania

L. V. BLACK: Mr. Schnure has brought up a number of points which are of vital interest to anyone connected with power distribution systems in steel mills. I was doubly interested in that section of the paper dealing with two parallel high tension lines, one a 6600 volt line, steel towers, and the other a 33,000 volt line belonging to the utility company on wooden poles and wooden cross arms, without a static or ground line protection. Just how free from surges and outages these two lines will be during storm seasons, is a point of much interest.

My experience goes back some years to a 6600 volt line at the Bethlehem plant across a section called the cinder dump. This line was constructed on wooden poles with wooden cross arms. The ground throughout this section is made up of open hearth and blast furnace slag with an average depth of 40 to 50 ft. so that the entire area is virtually a big insulator. With this construction very few storms passed by without some interference, many of which were outages.

We then rebuilt this line, using steel towers, steel cross arms, and a substantial static or ground line. Each tower was connected solidly to the ground line and the ground line was connected through a heavy copper conductor to adequate and substantial grounds at each end of the cinder dump as well as another ground buried on the bank of a creek at the half-way mark. Only one ground line was used, mounted at sufficient height above the conductor lines so that the lines fell within an angle of 40 degrees. Each tower was also equipped with a lightning rod extending 10 ft. above the ground line. Continuity of service and lack of interference from storms on this section since it was rebuilt some eight years ago has been very enviable, I can't

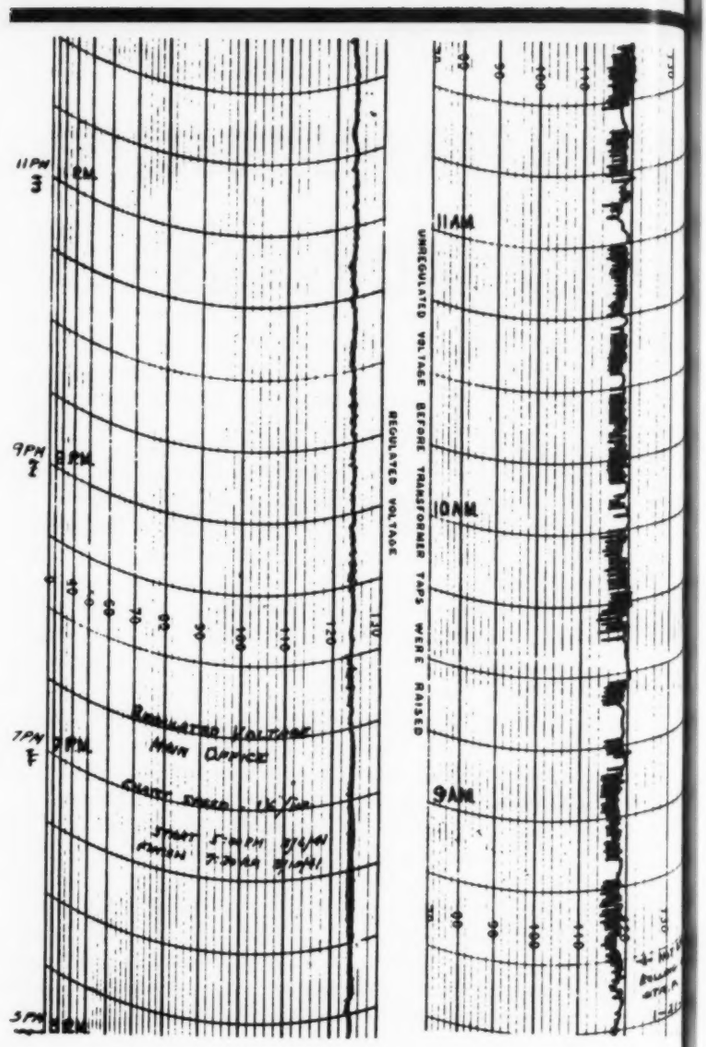


Figure 5—Charts showing conditions of voltage variation before and after installing the saturable core reactor.

recall of a single outage from storms on this section of line.

I might also state that all of our high tension lines are protected now with substantial ground lines.

G. O. VAN ARTSDALEN: One very interesting thing in this paper was the application of saturable core reactors on lightning circuits. This is my first intimation of their being used other than the way they were first brought out a year or more ago in the control of small heating furnaces. I think there are lots of possibilities with them, where close regulation is necessary.

J. C. REED: I might add some encouragement to Mr. Schnure on his installation of ground wires. It has been 23 years since we installed our 6600 volt system. That is, we put in 6600 volt generation and 6600 volt motors on the mills, and when that went in I covered every inch of the lines with ground wires, and in all that time I have never had a shot of lightning on the 6600 volt system, but have had lots of shots on the 2200 and 11,000 and 440 volt lines which are not covered.

Relation of Shock in HYDRAULIC LINES

By F. C. Schoen, Assistant Master Mechanic

THE MIDVALE COMPANY

NICETOWN, PHILADELPHIA, PENNSYLVANIA.

Presented before A.I.S.E. Philadelphia Section, April 5, 1941

▲ THE maintenance of hydraulic piping in connection with forging presses presents many interesting problems, and I am going to touch on one of these; namely, leaks and breakage due to shock, and what we have done to minimize delay and loss on each of two generally similar hydraulic forging presses, mentioning the trouble experienced and the methods used to correct the difficulty.

The first of the two presses is a single cylinder 42 in. diameter ram, four column unit, which develops 385, 1500 and 1800 tons pressure on the dies. These different pressures are obtained by using hydraulic water from 800 lb. and 2500 lb. hydraulic accumulator systems.

Operation of the tool is controlled, through a cam operated multiple spindle valve, known as the master valve, which in turn directs 2500 lb. per sq. in. water to the hydraulically operated forging valves. The cylinder of the press is prefilled with 800 lb. water which exerts the force of 385 tons. Next the 2500 lb. water is admitted to the cylinder exerting a die force of 1500 tons, a check valve preventing back flow to the 800 lb. system. It might be mentioned here that an easy check on the condition of the operating valves is possible, by observing the 800 lb. system accumulator, when it rises with the pumps not running, it is time to reseat them. A maximum pressure of 1800 tons is obtained by taking pressure off of the ram pullback cylinders while 2500 lb. water is applied to main ram.

Most of the trouble on piping here has been in connection with the 2500 lb. system. This high pressure water is brought to the forging valve from the accumulator through a 6 in. line approximately 150 ft. long, and all water exhausted from the press is supposed to return through an 8 in. line about 170 ft. long to an overhead suction water tank for the accumulator pumps.

When the press was first put into operation and the 2500 lb. pressure water flow to the cylinder was stopped by going die to die, pressure piping was loosened and in general everything was badly shaken up. Upon releasing the pressure, fittings were broken and joints loosened on the return or low pressure water piping.

Since each action was due to a different cause and both happened within seconds of each other, it was quite a problem to sort out the trouble. A study showed that from the time 2500 lb. water was admitted to the cylinder of the press and the ram went die to die, the 24 in. diameter hydraulic accumulator would drop from 6 to 7 ft., producing shock pressures over 3500 lb. per sq. in. To reduce this, a choke valve was installed in the main line of the press, then as the ram neared the end of its stroke it mechanically closed the valve, slowing down the speed of the water entering the cylinder. Heavy spring balanced shock valves were also installed in the hydraulic line, one at the press and one at the accumulator. I believe, however, that the biggest improvement was made recently when additional pumping capacity was installed. This gives more follow up of water in the system, and prevents extreme drops of accumulator, which in turn decreases the shock pressures. As another important cause of shock was felt to be due to the sudden release of stored energy due to tension in the columns of the press and a slight compression of the water, an attempt was made to bleed water off through a small line, before exhausting the high pressure water in cylinder. This helped a little but was not sufficient, most of the improvement seemed to be only in lessening the noise about the press. Perhaps it would have been more effective if the operators had allowed a longer time to elapse before opening the main exhaust valve. However, since this would slow up the tool a few seconds, it was not done. Relief valves for the return line were proposed, but it was felt that a valve of sufficient size would not respond as rapidly as was necessary and nothing was done. I believe that this might prove of some benefit and would be interested in knowing if anyone has tried valves of this type for a similar service.

The last and most successful attempt to reduce breakage in the return line consists simply of the installation of a ½ in. air line connected to works pressure and tapped into the return piping on the low pressure side of the press exhaust valve. This air serves to aerate the water and keep carrying it along to the suction water tank, so that when the high pressure is released from the press, it is practically the same as exhausting to atmosphere, instead of hitting a long column of quiet

water, breaking off ells and tees all along the line. As measured with ordinary Bourdon tube gauges, the pressure head of the quiet water is about 42 ft. and with the air jet in operation this does not appreciably change even when the press exhausts into return line. With no air being used the head changes from 40 ft. to about 125 ft., according to the gauge. However, this reading is, I believe, much lower than the peak pressures reached as the gauges do not respond too rapidly. It is possible that the breaking of fittings may have been due to the continued sharp hammer blows of high pressure water rather than pressures over 125 ft.

The second unit is a much larger press developing a maximum force of 5500 tons. This is also a four column single cylinder master valve operated type, but instead of its obtaining pressure from two accumulator systems, it uses a primary pressure of 1700 lb. per sq. in. and the increased pressure up to 5500 lb. per sq. in. is obtained by means of an intensifier using 1700 lb. water for power.

There have been and still are a considerable number of loosened joints and breaks in the hydraulic piping of this system. The original pipe sizes were in many cases much too large and gave practically no restriction to the flow of water. Various lines were reduced in size to as much as one-fourth the original area without changing the speed of the ram travel, and where this was done it is the unusual thing to have to renew a broken pipe or leaking joint. For instance, as originally installed a 4 in. double X pipe fed the pullback cylinders, pressure being controlled by a hydraulically operated valve. The pipe size was reduced to 2 in., the hydraulic valve removed and pressure control handled through two of the ports in the master valve. While there is still a noticeable jumping of the lines when pressure is turned on or off, very little trouble is experienced due to leaks. On occasion we have operated with a single port of the 1⅛ in. master valve and no appreciable slowing up of the press was evident. This is not a usual practice as we found extremely rapid cutting of valve and seat due to the high velocity of water, and it seemed that the best balance of maintenance time warranted the greater water pressure area, rather than continual reseating of valves.

The 1700 lb. per sq. in. water is admitted to the main cylinder by a hydraulically operated valve which is kept closed by 1700 lb. pressure on its operating cylinder. A necessity for accurate control of the ram means that this valve must close quite rapidly if over travel is to be prevented. Instead of using a choke valve as on the first press to slow up speed of the water, pipe sizes were decreased, and while this slowed up the ram speed slightly, it did not greatly affect the total production.

A third item which is still giving trouble is the operating valve for the intensifier power cylinder. This is controlled from the master valve and is what we call a two spindle valve. Each spindle is held closed by constant 1700 lb. pressure and is opened by another hydraulic cylinder. One spindle admits pressure water to the intensifier, the other exhausts this water. Some shock is experienced at the time the exhaust valve opens, and is, I believe, due to high pressure water hitting a column of rather slow moving water in the return piping. However, since the head to this line is low, little delay has been caused and an air jet, as previously mentioned, has not been permanently installed. The greatest trouble with this valve has been the breaking of the 1¼ in. columns holding the operating cylinders to the main valve body.

This can be explained due to the fact that the valves are rather carefully balanced and any tendency towards corrosion in the operating cylinders will cause spindles to stick and then suddenly slam open or shut. Close inspection with frequent cleaning of all moving parts has been the only way to decrease delays. An improvement may result by cutting down on the size of piping and valves, and we are now studying a re-design of the two spindle valve to reduce the main valve areas by 50 per cent. This should not slow up the speed of the press any as the intensifier is not used unless dies are on the metal being forged, but should materially reduce shocks in return water lines.

This partially covers the methods used on these two tools, and I believe it might be well to add a few construction details of the shock and master valves previously mentioned.

The larger shock valves as we use them, consist simply of square forgings inserted as tees in the pressure piping with a 4 to 6 in. diameter plunger passing through a gland and stuffing box. This plunger acts against springs which have sufficient initial loading so that under normal pressure they allow the plunger to compress them about 3 in. and with a 50 per cent increase in pressure due to shocks can allow the ram 5 to 6 in. more travel, the normal being 3 to 4 in. This, of course, means for a short stroke it is necessary to use springs with a high load constant and the valves do not respond too rapidly. A long stroke valve with soft springs may be an advantage, but after observing smaller valves with soft springs in operation where the beat of the accumulator pumps could be seen in the dancing of the shock valve, I think the more sluggish valve will be easier to maintain. Shock valves installed at the end of extremely long lines should be of benefit, though I have had no actual experience with any such installations and would be interested in learning just how much help they give others in shock reduction.

The master valves mentioned consist of a group of spindle or poppet type valves, one group for admitting pressure to the cylinders to be operated, the other group to release or exhaust this pressure. The spindles of the valves are raised and lowered by the action of cams on a shaft. Without going to very large diameter cams it is easily possible to control 5 or 6 different operations of a press, in a single revolution of the cam shaft. This in turn allows the operator more freedom in watching the work of the tool by cutting down the number of valves and motions he has to make and in addition definitely gives the proper sequence to all operation.

Once a valve is set up, very little leeway remains, and that only by adjustment of clearance between spindles and cams. This adjustment is primarily allowed so that valves can be reseated but smoother operation is sometimes obtained by careful adjustment of valve openings.

I believe a great deal can be done towards the reduction of shock but unless delays become quite frequent little attention is paid to causes of breakage and then it may be quite difficult to overcome. This, I believe,

is true in many maintenance problems and I hope that someone may benefit from these ideas we have used.

DISCUSSION

PRESENTED BY

G. O. VAN ARTSDALEN, *Superintendent of Maintenance*, Henry Disston and Sons, Inc., Philadelphia, Pennsylvania

F. C. SCHOEN, *Assistant Master Mechanic*, Midvale Company, Philadelphia, Pennsylvania

L. A. CAMEROTA, *Chief Engineer*, Florence Pipe Foundry and Machine Company, Florence, New Jersey

A. E. SWAIN, *Engineer*, Florence Pipe Foundry and Machine Company, Florence, New Jersey

A. T. HALSTEAD, *Assistant Mechanical Superintendent*, Bethlehem Steel Company, Sparrows Point, Maryland

G. O. VAN ARTSDALEN: I enjoyed Mr. Schoen's paper very much. When we talk of 1500 lb. pressure that put our little 600 lb. system far in the shade.

I notice all of Mr. Schoen's valves are remote controlled; using the primary water pressure as a control medium. I would like to know whether he has had any experience with any other control medium for their operation, such as solenoid, motor, or diaphragm operated. I would like to hear some experience from someone who has used such equipment.

F. C. SCHOEN: We have had little experience with anything but the hand-operated and remote-control valves, using hydraulic pressure on the primary pressure valves. I do not see any reason, though, why air or something else should not make a good control medium. I have been a bit afraid of electricity around the water end, because when a leak develops it might ruin the electrical equipment.

L. A. CAMEROTA: There are a few comments I would like to make generally. One thing in particular that I wanted to say is that I have observed in installing an electric system that you gentlemen are very careful about installing proper safety devices on each machine to which electric current happens to be applied. If the same principle were applied to hydraulic equipment, probably some of your troubles regarding hydraulic equipment would be eliminated the same as with your electrical equipment. No one, in other words, would think of putting a motor on any piece of equipment unless equipped with a circuit breaker or some other means for releasing the current in case of an overload. No one would think of throwing current across a line on a big motor. There would be something there to prevent you from so doing. Very frequently you will go in the plant and see a man yank on a lever of a hydarulic press and throw the full power into the press and expect it to function properly and to continue to function without undue maintenance and repair.

There is equipment available for taking care of these conditions in the same manner that you take care of them in your electrical equipment. I have often heard it said that engineers are poor salesmen and I kind of agree with that because there have been few engineers I ever heard speak that dwelt on anything else but facts and figures and nobody likes to listen to facts and figures. I don't propose to try to be a salesman up here but I do know our company has been doing some good work along this line.

I would like to add to Mr. Schoen's last statement about electrical control and air control. My favorite system for a remote controlled hydraulic valve is oil, using an oil pump, and an oil valve at low pressure.

I would just like to put a thought in your mind. About a year and a half ago we designed a 3000 ton press, a forging press—I won't mention where it is located—operating directly from an accumulator system using six air bottles at a pressure of 4500 lb. per sq. in. The stroke of the press is about 36 in. and we used as a means of operating the valve on this press an oil system. The spindles of the main operating valve are held on their seat by the 4500 lb. pressure. The pressure is applied by what we call the upside down system—the fluid enters under the seat and the pressure is therefore always against the seat of the valve. The pressure above the spindle acts the same as a heavy spring. A rocker mechanism actuated by an oil cylinder which in turn is controlled by a small follow-up valve raises the spindles off their seats. When the action is reversed the spindles come down on their seats against pressure. This will eliminate a great deal of the shock. As a matter of fact practically all of it is eliminated in this particular case. This press moves with such rapidity that you would think it was a steam hammer going down. I clocked it at about 30 in. per sec. I know very well we could get higher speeds than that if we had an operator who could stop it when he wanted to. We made various tests with the press and I don't like, as I say, to act as a salesman here—I am not trying to sell anything, but I am trying to give you facts—we have actually broken a crystal of a watch with this 3000 ton press. It responds instantly to the man's movement of a single lever. He can bring the ram down to within half an inch of an object or crack the crystal of a watch and go back again. This sounds almost incredible. We now have two of these presses in operation, at these speeds, without shock. I might add that these presses operate about 30 strokes a minute—30 power strokes—and as far as I know we have not had to change any parts in this particular equipment to date.

Again I say that if hydraulic equipment is given the same consideration as your electric equipment, much of the trouble that you may have experienced with hydraulic valves and machines can be eliminated in the same manner that your electric equipment troubles are eliminated.

A. E. SWAIN: I would like to ask one question in connection with the pressure coming up on the valves. We speak of not protecting the presses the same as with electrical equipment. What would happen when the accumulator pressure tends to raise the spindle off the seats?

L. A. CAMEROTA: The same pressure is used to hold the valve on its seat, the accumulator pressure. It

(*Please turn to page 40.*)

Some Improvements in Steel Plant

▲ A BACKWARD glance over the past forty years of maintenance work at Bethlehem Steel Company's Maryland plant reveals a number of changes and improvements in the design and maintenance of steel mill machinery.

Up until thirty years ago, it was the general rule for all bevel gear and most spur gear teeth to be cast, not cut as they are today. The babbitt bearings on line shafts were poured in place on the job. Most of the brass bearings on table rollers and jack shafts were used as cast, not machined, as in present day practice. At that time, fuel oil was a popular lubricant for table rollers and other bearings and it was also used as a coating on gear teeth. These are only a few instances in which present day practices show a great improvement over those of the past.

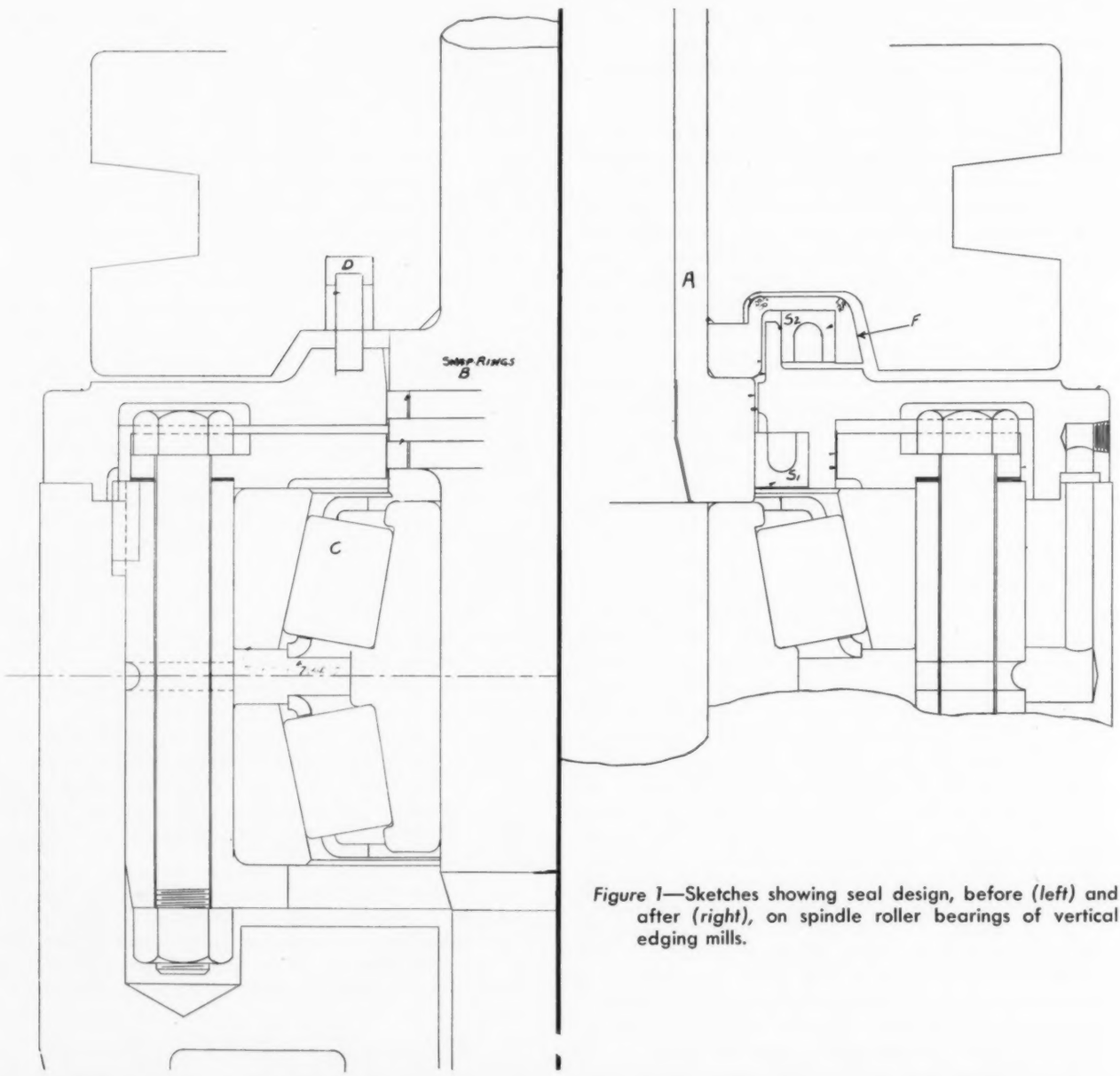


Figure 1—Sketches showing seal design, before *(left)* and after *(right)*, on spindle roller bearings of vertical edging mills.

MAINTENANCE *and* DESIGN

By A. H. Halstead

BETHLEHEM STEEL COMPANY

SPARROWS POINT, MARYLAND

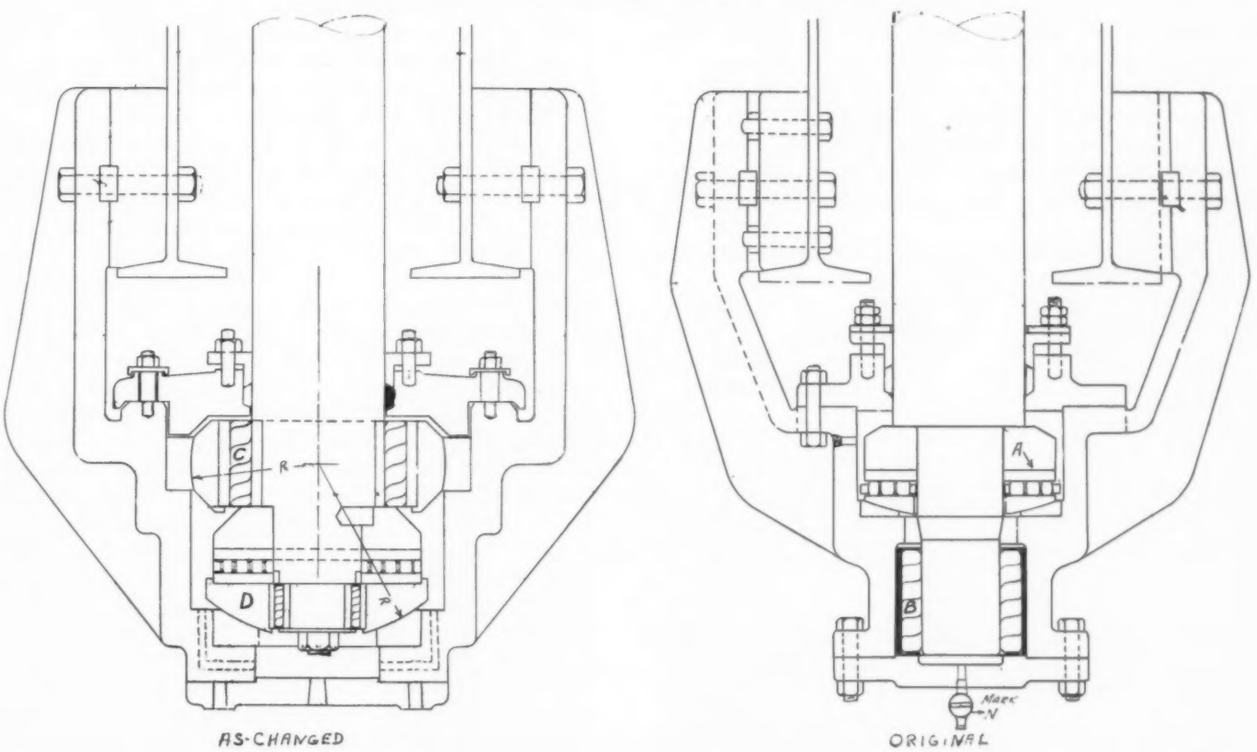
Presented before A.I.S.E.
Philadelphia Section, April 5, 1941

Today there is much closer coöperation between the engineering, operating and maintenance departments. The result is that many of the operating and maintenance problems which formerly were taken care of in the field, after the job had been installed, are now being eliminated on the drawing board. And by scheduling regular shut downs for routine repairs and inspection, the number of emergency shutdowns for breakdown repairs have been greatly reduced.

The following cases serve to illustrate some improvements in design made by the maintenance department which have eliminated frequently recurring trouble, and which may be of some value to men in other plants.

At the blooming and slabbing mills the ingots are carried from the soaking pits to the receiving end of the mill approach table in an ingot buggy. They are placed vertically in the buggy, and the pot on the buggy is tilted through 90 degrees toward the receiving table. Before the ingot reaches the horizontal position, the top part strikes against the first table roller which is of fluted design. Until the ingot reaches a completely horizontal position, so that the fluted roller can drag it out of the pot, the first roller just slips against it. As originally designed the rollers operated at 60 rpm. At this speed the fluted roller slipping against the ingot caused the whole table to chatter excessively. This heavy vibration resulted in roller wear and breakage, breakage of teeth in bevel gears on rollers and line shafts, and it kept the bearing caps on roller and line shafts loose most of the time. While operating at this speed, an average of five rollers and four line shafts were changed every year. By reducing the speed of the table

Figure 2—Sketches showing thrust bearing on the lower end of the vertical spindle of a flanging machine; right, original arrangement; left, as changed.



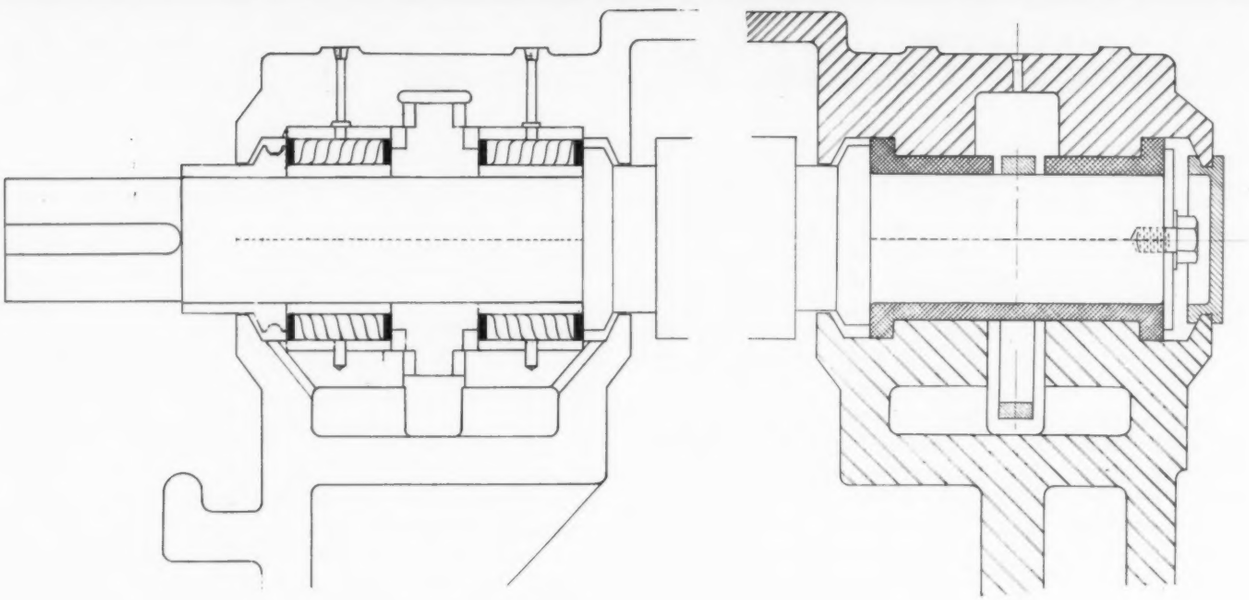
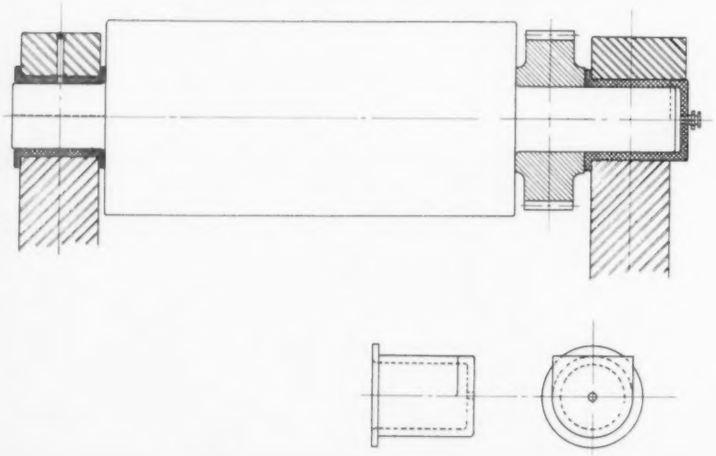


Figure 3—Replacement of brass bearing shells with roller bearings on pinion shaft increased bearing life from two months to four years.

rollers to 16 rpm., no line shafts and only three rollers have been changed in the past six years. Production has not been reduced as the ingot can still be delivered to the mill in plenty of time. Thus, maintenance cost was lowered and delays cut down to a minimum.

Another instance where reduction in the speed of a unit saved delays and lowered maintenance cost, with no decrease in production, was on the depressing table after the slab shears. This table has a swinging motion of about 4 ft. and it serves to drop crop ends down a chute onto a crop conveyor. Originally the drive consisted of a 75 hp., 440 rpm. motor connected directly to a worm shaft which meshed with a cast iron worm gear on a crank shaft. This shaft, in turn, was connected to a cross shaft with two segment gears, meshing into racks at the bottom of the table girders. During one year not less than three worm gears were worn out. A 50 hp. motor with back shaft was installed and the speed of the worm reduced to 90 rpm. A 15,000 lb. counterweight was added to the segment gears, as the weight had rested on the gear when the table was swung toward the shears. The present worm gear has been in service since 1935. Neither the rolling speed nor the speed of the shear was lowered by making the change.

Figure 4—The use of cup brasses *(right end)* is advantageous where water or mill scale are encountered.



A change in the design of the scale and water seal has proved very satisfactory in keeping water and scale out of the roller bearings on roll spindles of the vertical edging mills on the 21 in. continuous mill (see Figure 1). The section on the left hand side shows the original design, the section at right the improved seal. One trouble encountered with the original design was excessive wear on the spindles at the roller fit, due to the lateral movement in the rolls while the bar was in the mill. Due to the frequent change of these rolls it was necessary to give them a .015 in. clearance, to avoid difficulty in getting them off. Failure of top spindle roller bearing was another source of trouble (C) because the original design of the roller bearing seal (D) allowed water and scale to be forced into the bearing by the hydraulic sprays on the mill. The excessive spindle wear was eliminated by turning the spindle down and shrinking a hardened steel bushing at the roller fit. The original bearing seal consisted of snap rings set in grooves on the spindle, and a labyrinth. The changed design consisted of alteration in the cover plate to take seals (S-1) and (S-2), also a combination labyrinth and seal holder. A gasket was also put between the bottom face of the roller and the seal holder. Inspection of these bearings after two years of operation revealed no evi-

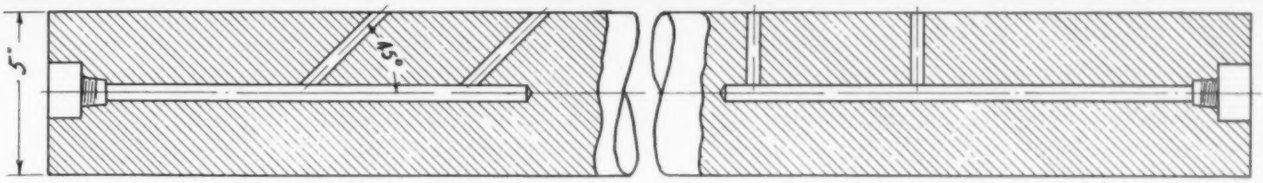


Figure 5—Ingot car axles drilled for lubrication as shown at right broke through oil holes. Drilling as at left eliminated breakage.

dence of scale or water getting into the bearings. As the life of the top roller bearings has been the major item of maintenance on these edging mills the outlined changes have proved well worth while. Figure 2 shows a thrust bearing on the lower end of the vertical spindle of a flanging machine. This spindle is subject to a downward pressure of 20 and 40 tons, while rotating at 46 to 92 rpm. It is also subject to lateral thrust, produced by the flanging roll pressing against the head being flanged.

The bearing at the top of the spindle is made up of four adjustable brass quarter boxes. When this bearing wears, or when the quarter boxes are not properly lined, the lateral thrust produced by the flanging roller forces the spindle out of the vertical line. The original bottom thrust bearing was designed to be self-aligning. However, self-alignment was not possible due to the application of the lower radial bearing below the thrust bearing. This gave the spindle a tendency to bend just above the lower radial bearing, thereby throwing the load on a small portion of one side of the thrust bearing. These bearings only lasted from two weeks to four months, depending upon the class and amount of work done. By placing the self-aligning radial bearing above the thrust bearing and mounting the former lower bearing in the upper half of the spherical shaped container for the thrust bearing, the load was distributed evenly over the entire thrust bearing surface. It is necessary that the radius of the spherical seat of the thrust bearing and the self-aligning bearing be struck from the same center. After this change the bearing lasted five years before wearing out.

By replacing brass bearing shells with roller bearings on the pinion shaft of a drive with 75 hp. motor running 625 rpm. (Figure 3) bearing life was increased from two months to four years. Running on brass shells this pinion shaft caused frequent delays as the shells wore rapidly and the shaft vibrated excessively. This gave the electrical department a good deal of trouble as the risers on the motors tended to break loose. These difficulties were eliminated by the installation of roller bearings.

Replacement of half brass bearing shells with cup brasses (or whole bushings with one end blank) is shown in Figure 4. Use of these bearings with grease connections in the solid end is desirable, particularly adjacent to mills or shears or in other localities where water and scale are encountered. The lubricant has to flow through the whole length of the journal to get out the open end, and may carry along a considerable amount of dirt and water. About half a year ago this type of bearing was installed on the front end of a shear dip table. Using half bearing shells with both ends open, it was necessary to overhaul the first three rollers and the idle shaft every two weeks—a job calling for two men. Since the change was made no repairs have been necessary on this section of the table.

In one mill an 18 in. accumulator on the shear pull back cylinders proved to be a constant expense both on account of the time lost in shut downs and the materials needed for repairs. When this shear was duplicated in another mill, a 26 in. accumulator was installed. This has been in operation for nearly five years, and has never needed any repairs. Installation of sufficiently large accumulators is, therefore, good economy.

Originally the axles on the ingot mould cars were lubricated through the bearing housings. As the bearings were on the inner side of the wheels, they were unhandy to reach and did not receive proper lubrication, causing excessive axle wear and damaged bearings. Figure 5 shows how the method of lubricating was changed by drilling ½ in. pipe tap holes in the end of the axle and ⅜ in. holes from the journal into the center hole. This offered an easy way of lubricating and the bearings lasted longer. However, soon the axles started to break through the ⅜ in. oil hole. Upon the suggestion of one of the foremen, these holes were then drilled on a 45 degree angle, instead of at a right angle with the center line of the axle. This proved effective as not a single axle drilled that way has been broken.

DISCUSSION

PRESENTED BY

N. C. BYE, *Chief Engineer*, Henry Disston and Sons, Inc., Philadelphia, Pennsylvania
L. A. CAMEROTA, *Chief Engineer*, Florence Pipe Foundry and Machine Company, Florence, New Jersey
H. K. HARDCASTLE, *District Manager*, Electric Controller and Manufacturing Company, Philadelphia, Pennsylvania

N. C. BYE: The thing that impressed me about that paper is not only that Mr. Halstead has been able to accomplish a great deal in correcting his trouble in maintenance, but it is a thing I have found a great deal in almost every piece of equipment, that is, that without

exception, almost no matter where you buy the machine, there is always something you have to do to it to be able to run it satisfactorily for any length of time. It seems to me a shame that the manufacturers and representatives do not follow up more. This is maybe a severe criticism but it seems to me a shame they do not follow up the details of the maintenance man on his problems. I don't know whether it is because the users of the machines do not go back to the manufacturer enough to see that these items are corrected not only in their own plant but in other places, or whether it is just overlooked and forgotten in each individual plant.

We just bought some furnace equipment which is a duplicate of material which we have had before. We went through a period of getting it in operation on the previous machines, the previous work. We had written up a long list of things which we thought could be corrected on the equipment that was just being purchased. Those items were the result of our experience on the previous jobs, and rather to our surprise we find a lot of the same things are duplicated in the new machines. I don't know whether it is because the manufacturers are a little loathe to change their designs or drawings to correct these details, and they are minor details—but they mean a lot of work to the maintenance man to correct them.

So I hope the points Mr. Halstead brings out serve not only to show what can be done in eliminating maintenance troubles but impress on the manufacturers the fact that they can learn a lot on how to move the equipment from the maintenance man's angle as well as from the requirements of new processes.

L. A. CAMEROTA: I enjoyed Mr. Halstead's paper very much—it shows marked ingenuity on his part. I also want to emphasize, as a manufacturer, the statement that Mr. Bye made. It is a fact that very often improvements are made by the various maintenance men, the men in charge of maintaining the equipment—whether they be engineers or just mechanics. Very often these improvements are not recorded or the manufacturer of that piece of equipment doesn't have sufficient personnel to follow up their equipment, and the trade in general loses the benefit of the improvement that has been made under the circumstances. It is very gratifying to see men of this type, sharing their ideas and exchanging their thoughts and ideas and bringing out these points so that we who have to do with the designing of equipment of that nature can benefit by it.

H. K. HARDCASTLE: I just wanted to congratulate Mr. Bye on his remarks about developments in equipment due to service requirements. However, I do not think the manufacturers are always to blame for not keeping posted on all details, because under the present conditions there are some occasions where we have installed new types of equipment or made changes and have honestly tried to follow them up, but found it awfully hard to get past the front office. We are really delighted to get our noses into the dirt. We really want to know the inside dope, but it is sometimes difficult to get. I know in our case, and I am sure it is also so in the case of other manufacturers, that the present designs are due to troubles and corrections that have been made to suit the users, and that is the way we have progressed. When a representative calls on your plant, it does not necessarily mean that he came to sell a wooden nutmeg to the purchasing agent.

Schoen Discussion

(Continued from page 35)

passes under the seat up through the side of the valve to the hold down cylinders.

The oil pressure is only used to actuate the control rocker under the valve.

A. T. HALSTEAD: We do not have as long high pressure lines as Mr. Schoen has. All our high pressure lines over 700 lb. pressure only run from intensifier to shears on which we will occasionally blow a joint. On our hydraulic lines through our mills, we only carry 700 lb. pressure. We do get considerable shock on these lines, particularly where we have large operating valves.

We use air bottles and spring bushings to relieve this shock. Air bottles make a good shock absorber, providing you keep them charged with air. We first tried charging them by putting a connection in top of bottle, but found that air would leak out, causing bottle to become filled with water and several of them split. We now make them with a connection in neck of bottle and recharge them every few weeks with 100 lb. air pressure. Since doing this, we are getting very good results.

Maintenance KINKS and GADGETS

By Walter H. Burr

Superintendent Electrical Department

LUKENS STEEL COMPANY

COATESVILLE, PENNSYLVANIA

Presented before A.I.S.E.
Philadelphia Section, April 5, 1941

▲ A BRIEF outline of the organization and method used in handling the maintenance of electrical equipment at Lukens may give you a better understanding of the problems involved. The electrical department hire all crane operators and assign them to their jobs but the operators take orders from the department in which they work. This, we feel, helps in keeping down the cost of maintaining the cranes. Complete repairs to cranes are handled by the electrical department. Each producing unit has its own maintenance group. An effort is made to take care of inspection and repairs on day turn. Some routine inspection and lubrication is taken care of on the other turns. In most of the departments one man on each night turn looks after the work.

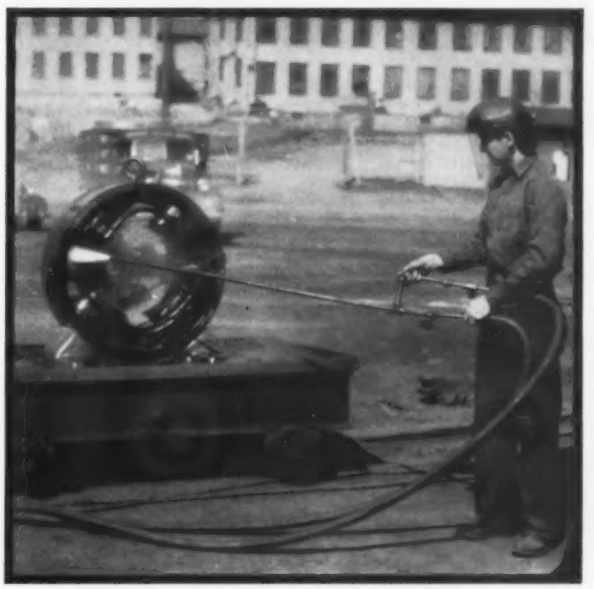


Figure 2—View showing cleaning operation on motor frame, using spray gun.

Figure 1—Showing motor dismantled ready to be cleaned.

Only field repairs on electrical equipment, as a rule, are taken care of by the maintenance groups in the departments. In many cases a change of the complete motor in trouble results in less delay than the changing of parts. Motors are assembled and tested in the shop and in most cases are placed in a central motor storage room ready for use. If delays are serious, a spare motor is carried in the department ready for use.

CLEANING AND DRYING OUTFIT

If trouble develops on a motor, the maintenance group make the change and deliver the motor to the electrical shop. Here it is dismantled, placed on a truck, taken outside the shop and cleaned by means of a spray gun using cleaning spirits. In some cases the fields are removed, in others it may not be necessary.

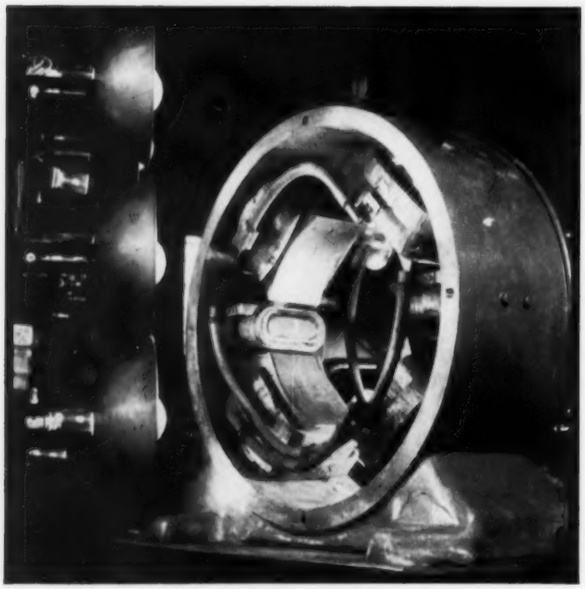
The type of spirits now used does not dry very rapidly so some method of drying was considered advisable especially if the equipment is urgently needed. A steam heated oven in the basement of the shop was used on certain occasions but because of the handling involved, it was not used as much as it should have been.

About four months ago we decided to get a couple of

Figure 3—Showing motor frame after cleaning.

infra-red ray lamps to see what they would do. I gave them to the man who handles motor repairs. He was so favorably impressed with the short test which he had made that we purchased a couple dozen of these lamps and in a very short time I noticed he had rigged up

Figure 4—A battery of intra-red lamps are used for drying cut the motor frame.



stands for these lamps which were suitable for the particular jobs for which he wanted to use them.

The most frequent application made of these lamps is that of drying out equipment which has been brought in for cleaning, overhauling and painting. Only recently I find they are being used to heat up the windings on armatures before they are banded.

This heating equipment we believe is going to be very useful in many operations performed in the electrical repair shop.

SMALL POWER HAMMER

Only recently a small tool has been added to those used in the electrical shops which has become very popular. It is a small portable electrically driven power hammer. Briefly it consists of a small motor driving a flexible shaft which in turn actuates a plunger pump which compresses air against the head of the tool being used, imparting some 3500 impulses per minute. By varying the pressure applied to the tool you vary the force exerted on the cutting tool.

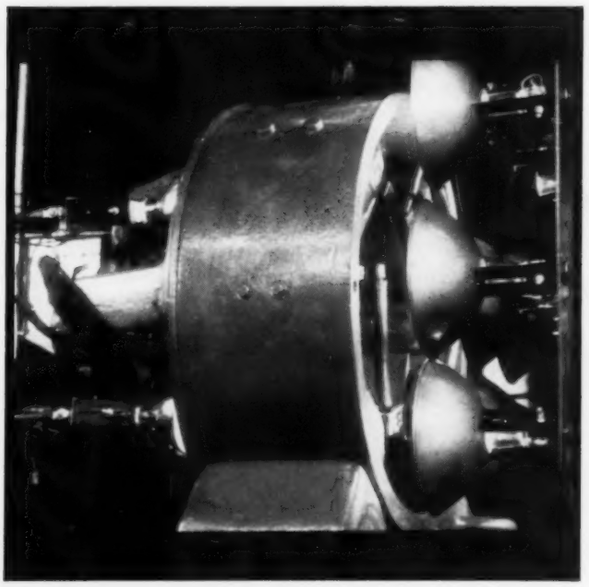


Figure 5—View showing battery of lamps in position on one side of motor.

For cutting oil grooves in bearings it is very effective. By proper use these grooves can be cut without changing the contour of the bearing and they can be cut in very much less time than by hand.

For cleaning up keyways or cutting small keyways it is unusually effective.

MAGNET REPAIRS

We have in service some 75 magnets of various types and sizes. Two of these are possibly of unusual interest. They were designed and built by my predecessor, F. H. Woodhull, and have demonstrated such high lifting

capacity for their external dimensions that they have never been replaced by more modern design. They are used to load the very thick heavy sections which are rolled on the 206 in. mill; also to load the very wide plates in cars. A single cable is fastened to the center of the magnet which helps in loading the very wide plates. Part of our experience on magnets was gained while repairing these special magnets.

We do not rewind the large magnets in our shop but we do make all other repairs. We have found that some of our trouble with rectangular magnets was caused by the terminal leads from the coil where they cross over the winding before they entered the terminal box. To overcome this on rectangular magnets we have changed the location of the terminals so that the wires leaving the coil come directly into the terminal boxes. This trouble may be due to the severe use to which these magnets are subjected in our plant. We, at least, think that we have eliminated some of our troubles by making this change.

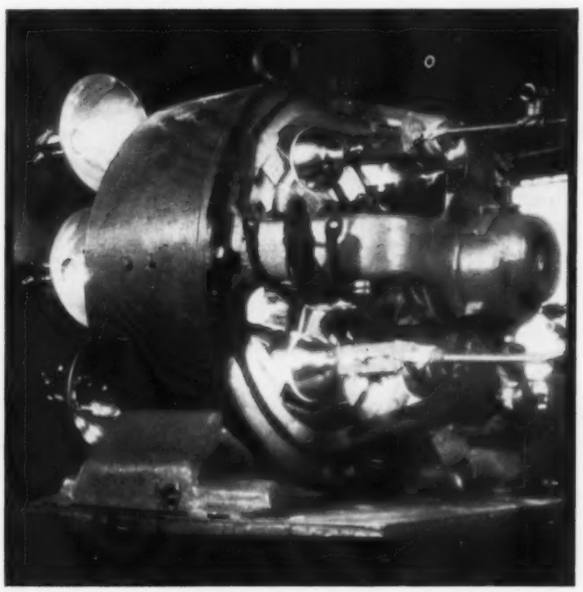


Figure 6—Showing lamps applied to both sides of motor frame, ready to start drying.

Figure 7—On this magnet coil, the leads come directly out of the winding.

INSULATING MATERIAL

In connection with the work of the Electrical Developments Committee of the A.I.S.E., I have watched with interest the progress made in the use of fibre glass insulation. I tried not to become too enthusiastic about it but we have used it on many applications.

It has been the policy at Lukens for years to buy the coils for the various makes of motors from the builders and we rewind most of the armatures in our shop. We wind some wire wound field coils, some magnet coils and some brake coils. We have used for this purpose double and triple formex wire, double glass and a combination of glass and formex insulated wire. Results obtained have been very satisfactory.

We are now ordering practically all our armature coils with mica glass insulation. It has been nearly three years since we started to use them and we have lost only three mica glass windings during this time. One was due to bands coming off and the others due to vibration which caused these coils to ground in the slots.

Now due to the reduced section required by the glass and mica insulation we are using oversized copper coils wherever possible, which still further reduces the failures on applications where motors were frequently overloaded. Slot size coils we feel to be another step toward reduction in maintenance costs.

INSULATION TESTS

For a number of years we have been keeping records on insulation values of certain pieces of electrical equipment, principally main drive units. We have found these of so much value that we have recently purchased additional meggers in order to make it possible for the foremen in the various mills to test out all of the equipment which is essential to the continuous operation of these mills at frequent intervals.

It does two things. It has taught all these men what dirt and oil do to the insulation of fields and armatures and the wiring in their own departments and it enables them to locate and change motors over the week end when the mills are down on which they cannot get the proper insulation resistance readings.

We try to keep the insulation readings on the various 230 volt d-c. mill type motors above 200,000 ohms and when readings under this value are obtained, they are returned to the shop for overhauling and cleaning. Con-

ditions under which tests are made materially affect the readings.

In addition to periodic inspection whenever one of the main drive motor units is cleaned or overhauled, a record of the insulation resistance is taken and these are plotted on the insulation card on file in the office for that machine.

I do not know what your experience has been but we think lifting magnet circuits cause more damage to insulation on d-c. equipment than possibly anything else in the steel plant and for that reason make a special effort to keep these magnet controls and connection in proper condition.

SPECIAL FAN-COOLED MOTORS

In the open hearth pit at Lukens we have some unusual conditions due entirely to the fact that we bottom pour most of our ingots and roll a large proportion of our ingots directly into plates. We have as a result a very wide range of mould sizes and that to a large extent limits us to the use of ordinary overhead electric traveling cranes in the pits. All the stripping is done in the pits with these cranes. The dirt floor in the pits is leveled off quite frequently and to do this a heavy mold is dragged up and down the pit with the crane. This is some job for a bridge motor and we have had plenty of trouble due to burnouts, etc. We finally purchased a small ventilating fan driven by a ¼ hp. motor, delivering 900 cu. ft. per min., mounted it on the cover to the MC motors and whenever the crane is in service, this fan is forcing air through the motor and out the back end through cover plates.

We had some trouble with the small 230 volt d-c. motors on these fans at first and are now using 110 volt motors and putting two sections of resistance in series with this motor, one-half on the positive side and one-half on the negative side. The burn-outs on bridge motors have been almost eliminated and with the resistance in circuit with the 110 volt fan motors, the failures in these have been greatly reduced. On some of the mill type motors on other than crane applications this scheme has been of real value but in most of these cases 110 volt a-c. motor driven fans are used.

Figure 8—Alteration of magnet case, as here shown, permits bringing leads straight from coil through frame.

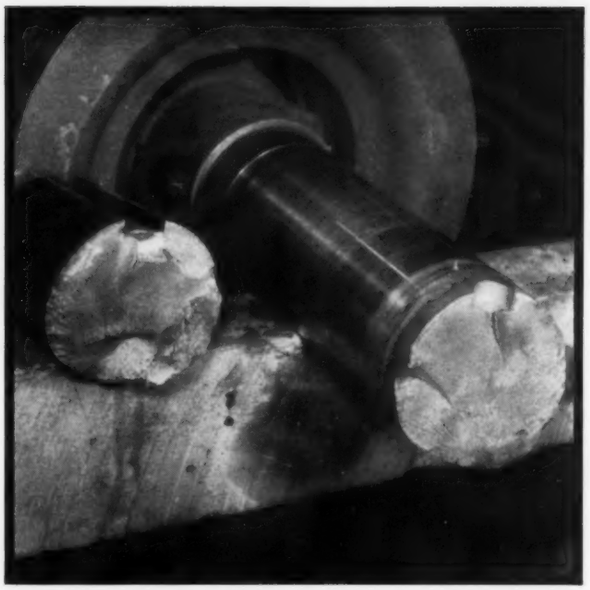


Figure 9—This armature shaft failed after welding up keyway and building up on taper.

COMMUTATOR GRINDING

As a result of several years' experience on grinding commutators on the larger d-c. motors and generators we are doing certain things which may be of interest to some of you.

Copper and abrasive dust can be the source of a lot of trouble especially when deposited in back of the risers and in the armature slots and pole face windings on large d-c. motors. To prevent this we now use a very heavy paper and make a hood over the whole front of the armature enclosing the risers and extending down to the commutator bars themselves. This is fastened to the risers with glue. Sometimes it is necessary to put a tape band around the armature just back of the risers which helps to hold this paper in place. Where considerable copper is to be removed, a suction fan is used, placed near the grinding and exhausting to the outside atmosphere.

In order to grind the full width of the commutator at one setting we use a stationary motor direct connected to a long pulley. As the stone is fed across the commutator, the belt travels across the pulley on the driving motor.

It is very essential that the mica on slotted com-

mutators be kept below the commutator surface, and after slotting, a V-shaped file is used to remove all remaining mica. Many cases of commutator trouble can be corrected before they become serious if the commutator is sanded with a large stone made especially for that purpose. Better results are usually obtained if the commutator can be ground at full speed. Conditions are such that this is not always possible.

We use an old lathe chuck on which a six-prong spider is welded to grip the shaft of rotaries or other units which provides a convenience of turning the armature by hand when slotting commutators.

WELDING ARMATURE SHAFTS

Driving the approach table on the 206 in. mill we use two 100 hp. motors. Some time ago someone did not follow up on the keys properly and the keyway in the arm shaft was damaged. It seemed like needless expense to put in a new shaft so we welded up the keyway and also welded up the taper. A new keyway was put in the shaft on the quarter and the taper brought up to standard size. Care was taken to keep the heating of the shaft down as low as possible. We were not equipped to stress relieve this shaft in the armature. It looked like a good job and was put in service but about three weeks ago on night turn it broke off just beyond the point where the welding stopped. Result, a delay on the mill when its production was important.

You can see from the illustration what had taken place. If you weld armature shafts, you must stress relieve.

Figure 10—Close-up view of fractured armature shaft shown in Figure 9.

206 IN. MILL COMMUNICATION SYSTEM

The 206 in. plate mill at Lukens Steel Company as it is known to the employees consists really of two mills. The 140 in. mill, a 3-high steam driven unit with its own heating section, layout floor, and shearing section and shipping department. Parallel to it is the 206 in. 4-high reversing steam driven mill with its own heating section, floor section, shearing, gas cutting department and shipping department. The heating sections are arranged to serve either mill by simply transferring ingots from one to the other on a transfer car.

The dimensions of this 206 in. mill, itself, and the auxiliary equipment are such that the various operators never could communicate with each other. This condition is made still worse by the noise which results from the use of salt on the ingots as they are being rolled into plates.

Due to the physical dimensions of the 15,000 hp. reversing engine driving this 206 in. mill, the engineer is located on one side of the mill and the screwman, table operators and roller are on the opposite side. The screwman and table operator are in an elevated operators' pulpit and the roller is on the floor level at the rolls.

How to get closer coördination between these various operators has been a serious problem. About three months ago we decided to try out a communication system taking advantage of modern methods of applification to overcome the noise of the mill. Tests were made to determine the amount of amplification necessary. No effort is made, however, to overcome the noise caused by the explosion when salt on the plate hits the mill.

The roller, engineer, screwman and table operator are each provided with a microphone and a loud speaker. On the floor of the heating sections of both mills are microphones and loud speakers so located that communication can be carried on between all of the operators on this mill at all times. When the roller wants to give special instructions regarding certain ingots, he pushes a button on his mike, issues his instructions, and can be heard by all the men in this crew.

Due to the fact that the engineer uses both hands in his regular operation a foot operated switch allows him to talk to the group at will. The switches on the mikes prevent the amplification of the noises around the operator who is talking except during that interval while he closes the switch on his mike. Naturally, in the heating section it requires several amplifiers to enable the heater in any part of the heating section to hear at all times.

Although it has been in service only a comparatively short time, we believe we have found the answer to a very difficult problem.

CONCLUSION

As I stated at the beginning the purpose of this paper is to bring out a discussion of the problems which you have, so tell us how you handle your maintenance problems. The phenomenal progress in the steel industry is due in no small measure to the fact that frank and free discussions of the problems involved have taken place between the representatives of the various steel companies.

DISCUSSION

PRESENTED BY

N. C. BYE, *Chief Engineer,* Henry Disston and Sons, Inc., Philadelphia, Pennsylvania
W. H. BURR, *Superintendent,* Electrical Department, Lukens Steel Company, Coatesville, Pennsylvania
H. H. ANGEL, Electrical Department, Bethlehem Steel Company, Sparrows Point, Maryland
F. C. SCHOEN, *Assistant Master Mechanic,* Midvale Company, Philadelphia, Pennsylvania
JOHN C. REED, *Electrical Engineer,* Bethlehem Steel Company, Steelton, Pennsylvania
L. V. BLACK, *Superintendent,* Electrical Department, Bethlehem Steel Company, Bethlehem, Pennsylvania

N. C. BYE: There is only one thing I wanted to ask: that is, what solution do you use on the spraying?

W. H. BURR: Solvent spirits. It is used with an atomizer connected to a compressed air line.

N. C. BYE: Is that the only thing you ever have used? Do you ever use any combination of naphthas or carbon tetrachloride?

W. H. BURR: At one time we used carbon tetrachloride but now use only the spirits. In connection with the heating with infra-red ray lamps, let me say it seems to bring out any oil that may be in the laminations or in the windings more effectively than any other method which we have used.

N. C. BYE: Do you pre-heat?

W. H. BURR: No.

H. H. ANGEL: We have often wondered just about how far to go in measuring insulation of the ordinary mill motor to determine when it should be changed. We do that periodically with the main drive equipment, but we have never done it with the ordinary table or mill auxiliary drive and are not convinced that it is very practical, but there is plenty of room for thought on such a topic.

W. H. BURR: I might add this: When you get repairmen in the habit of checking the insulation, you will be surprised how clean they keep the motors as compared with what they were before this practice was started.

F. C. SCHOEN: In connection with the welding of the armatures and stress-relieving, I wonder if it is practical to use high-frequency electric power.

W. H. BURR: This method of stress-relieving has been used but it would take special equipment which is not usually carried in an electrical repair shop.

F. C. SCHOEN: In connection with the solvent spirits such as Mr. Burr uses for cleaning motors, we have used same for cleaning the cloth elements of lubricating oil filters and find it does a good job and is very safe around the engine rooms.

J. C. REED: I noticed Mr. Burr has had some failures on his armature shafts, due to welding. He is perfectly correct in his statement that if you want to weld any kind of shaft steel you have to heat it afterwards, if you want to avoid breakages. We do not weld any armature shafts; we do all our building up of journals with a metal spray process. We have found that to be very successful and very satisfactory. I don't know whether there is anybody selling metal spray guns around here or not, if so, they can probably tell you about it. We have not only been using this process quite successfully, in connection with armature shafts, but have sprayed successfully gas engine piston rods, and they are better than the original rods. I would suggest, therefore, that anybody having to do with the maintenance of shafts and journals, might look into the process. It is well worth-while.

As to magnets, we have a complete magnet repair outfit. We rewind them on a table similar to a boring mill, that is, in a flat position, and we can rebuild magnets, no matter what the trouble. There is nothing to stop us except the patents, if any, from building a complete magnet. It's something like the old sailor who had a knife. He said he had had it for 50 years but had put in seven new handles and five new blades.

L. V. BLACK: There seems to be quite a little interest in Mr. Burr's paper dealing with welded armature shafts, and I see no reason why we should get unduly alarmed about breakages. True, a welded shaft is not as good as a new and unwelded shaft, but at the Bethlehem plant I dare say there are hundreds of welded armature shafts in service on cranes, mill auxiliaries and machine tools, and producing good results. We have been welding armature shafts for the past 15 years and are continuing to do so. None of the shafts are stress-relieved, as it is impossible to do this while the shafts are still in the armatures. I also admit that occasionally we have a welded shaft break. Metallurgical departments do not recommend the welding of armature shafts.

The problem to be considered in the welding of armature shafts is an economic one. The cost of a new shaft plus the cost of stripping and rewinding an armature must be balanced against the cost of welding and remachining the old shaft. Or on the other hand, if a new shaft is not immediately available, is it better to weld the shaft or let the motor out of service until a new shaft is secured?

Our records have proven that many of our welded armature shafts have been in active service for periods as long as five and six years after welding.

Another matter to be considered in the welding of shafts is their size. It is very questionable as to the economy of welding small diameter shafts. They should be metal sprayed as described by Mr. Reed. We have brought up to size many small armature shafts by the metal laying process, and have had great success with this process. But here again damaged keyways cannot be corrected with this method, faulty keyways must be corrected by welding. There is insufficient strength of the sprayed metal to withstand the action of the key and as a result the metal will break away adjacent to the keyway.

W. H. BURR: The deciding factor then is how serious is the delay when the welded shaft fails.

L. V. BLACK: Right.

MEMBER: Do you use a vacuum pump in repairing magnets?

W. H. BURR: We do not. At one time we did put the compound in the magnets under slight pressure making use of several vent holes but this practice has been discontinued.

Report of CRANE GIRDER TESTS

By I. Madsen

Assistant Research Engineer

Fritz Engineering Laboratories

LEHIGH UNIVERSITY

BETHLEHEM, PENNSYLVANIA

These reports are based on tests forming a part of a research program, sponsored and financed by the Association of Iron and Steel Engineers, in connection with the revision of standard specifications for mill cranes. The work was carried out at Fritz Engineering Laboratory of Lehigh University under the direction of Bruce Johnston, associate director of the laboratory, and in direct cooperation with the Crane Specifications Committee, F. W. Cramer, chairman.

The results of the investigation will be incorporated in the revised crane specifications which will be available early in 1942.

1. Lateral Load Tests on Box Girder Mill Cranes . . . .

▲ THIS REPORT summarizes the results of lateral load tests on the bridge girders of three steel mill cranes. The first crane, No. 4683, was a 10 ton riveted crane of 88 ft. 4 in. span. The girders of this crane were of uniform box section throughout and notched at their ends to fit over the end carriage. The girders were bolted to the end carriage. The second crane, No. 8441, was a 30 ton all-welded crane of 104 ft. span and was uniform in section for 92 ft. The girders rode on four two-wheel equalizer trucks, and were tied together by a box end tie. The third crane was a 60 ton riveted, fish-belly type of 120 ft. span, No. 5886. It also was supported on four two-wheel equalizer trucks, and the girders were tied together by a box end tie.

The girders were loaded laterally on the top flange to simulate the manner in which acceleration and deceleration forces load the girder. The load was applied to the top flange by tightening a turnbuckle fastened to the cover plates of both girders. It was found that the whole cross section of the girders resisted the lateral load and not only the top flange as is usually assumed. In addition, the end ties and gusset plates of the end trucks caused the whole unit to act as a frame so that partial fixity was present. The end moments at the ends of the bridges varied from 60 to 84 per cent of the fully fixed end moments.

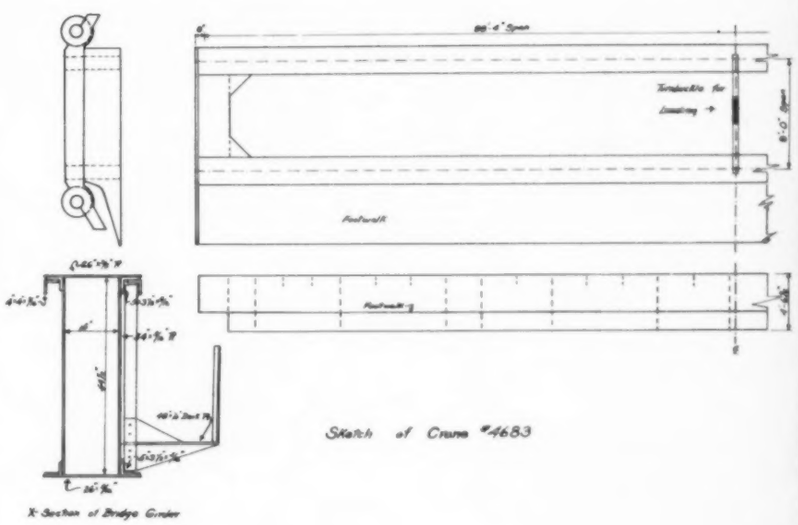


Figure 1

INTRODUCTION

These tests were made to determine how much of the section of a box girder is effective in resisting the lateral loads. The large resistance of box girders to twisting would indicate that the whole section of the girders resists lateral bending and not only the top flange as is usually assumed. To determine this, tests were made on cranes in the fabricating shop to determine the stress distribution in the girder. Knowing the stresses acting, one can compute what proportion of the cross section of the girder resists the lateral load. In the shop tests, the end ties were bolted to the girders and the whole assembly was supported on the equalizer truck pins on cranes No. 8441 and No. 5886, and on the end truck for crane No. 4683. Sketches showing the details of the girders are shown in Figures 1, 9, and 20.

The stresses were measured by means of a 10 in. Whittemore strain gauge. Gauge lines are established at the various sections of the girder where the stresses are desired by drilling two very small holes in the girder 10 in. apart. The distance between the holes is measured with the gauge before and after the load is applied. The gauge measures to 0.0001 in. The difference in length

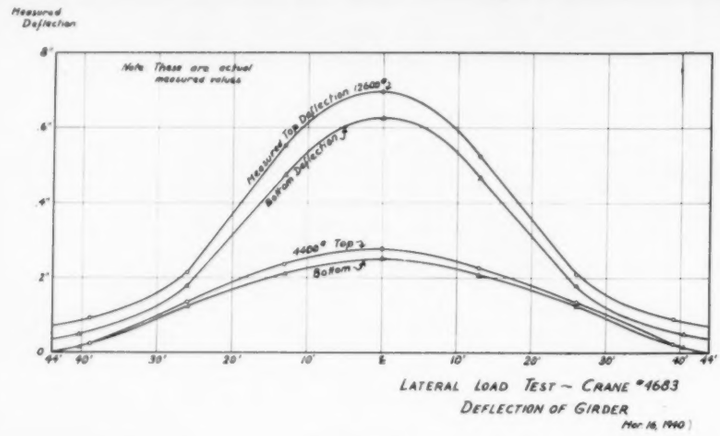


Figure 2

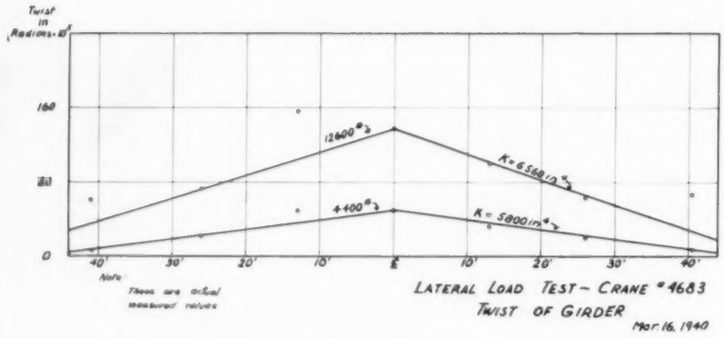
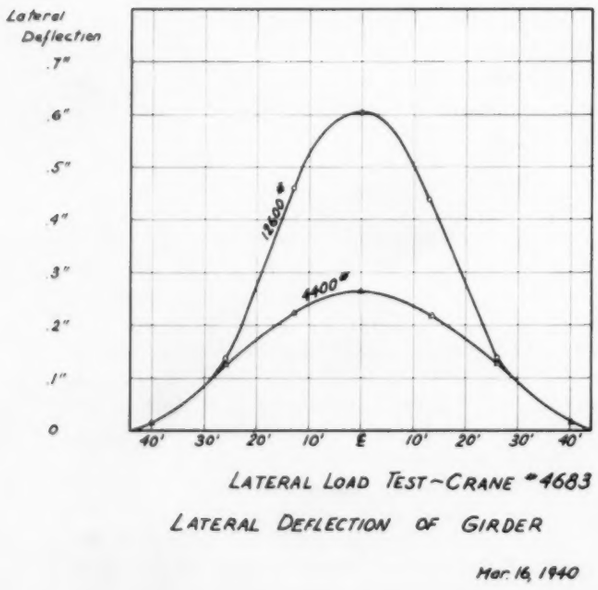


Figure 3

▲

Figure 4



between these two readings gives the strain in the member due to the applied load. The results are generally accurate to plus or minus 300 lb. per sq. in.

The lateral deflection of the girders was found by measuring the change in distance between the adjacent flanges of the two girders. This was done on the top and bottom flanges at seven sections using a 0.001 in. Federal dial. The difference between the readings of the zero load and final load gave the total movement of both flanges. The difference between the lateral deflection of the top flange and bottom flange at the same section divided by the distance between the flange reading points gave the twist of the girder at that section in radian units. Since the twist curves are computed from the difference of measured differences, the plotted points for the twist curves show some scatter. Knowing the twist, one can determine the average lateral deflection of the girder.

The load was applied by tightening a turnbuckle fastened to the two girders. A calibrated spring was interposed in the system and its closure was measured by 0.001 in. Ames dials. The closure was a measure of the actual load. This loading simulated the manner in which the braking and accelerating forces load an actual crane, except that in the actual crane both girders are deflected in the same direction, and in the mill tests the girders were bent in opposite directions.

TEST RESULTS

Crane No. 4683—The deflections as measured were the combined deflections of both girders. In Figure 2 are plotted the average deflection of each girder or the total measured deflection divided by 2. These deflections are plotted along the length of the girder and the center line mark denotes the center of the girder span. It should be noted that the deflection at the end tie is not zero for the 12,600 lb. load but is about 0.07 in. at the top flange and 0.035 in. for the bottom flange due to slip in the end connection.

The measured twist of the girder is shown in Figure 3. Some initial slip is also present. The torsional constants K, as computed from the curves, are marked on the corresponding curves. They are fairly close to the theoretical value of 6300 in.^{4}, as computed from Bredt's theory.*

The lateral deflection corrected for the initial slip noted in Figure 2, and also for the twist shown in Figure 3, is plotted in Figure 4. The slip correction was made by adding or subtracting a value to the measured deflection of the top flange so that the deflection of the end would be zero. This was also done for the bottom flange. These corrections were only a small percentage of the total deflection and are made because the elastic twist and deflections at the ends of the girder are zero. The curves for the different loads have a somewhat different type of curvature due to the greater end fixity of the girders when the 12,600 lb. load was on the girder.

The stresses were measured on the idler girder. Readings were taken on seven sections along the girder. At

*Rudolph Bredt. *Kritische Bemerkungen zur Drehungselastizitat Zeitschrift des Vereines deutscher Ingenieure*, July 1896, p. 785.

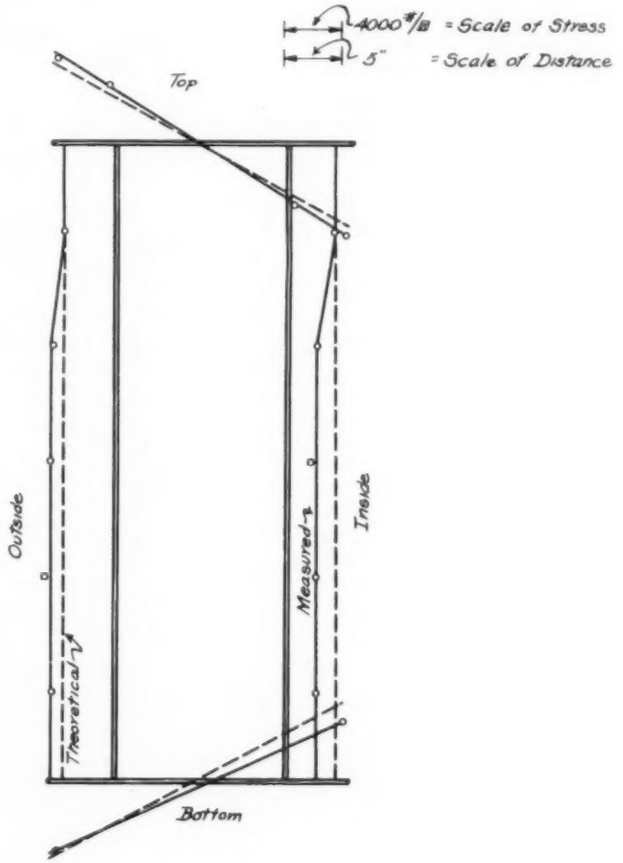
each section a number of readings were taken. On the top cover plate, readings were taken at the edge and 5 in. from the edge. They were also taken at the center line of the web and at distances 10 and 20 in. above and below the center line. Readings were also taken on the bottom flange angle 1 in. in from the edge of the angle.

Figure 5 shows the stress distribution at the center section and also shows the location of the gauge lines. The stress is plotted from the adjacent side of the box section as an origin.

In Figure 6 is shown the variation of the stress along the girder on the inside and outside edges of the top cover plate. The corresponding stresses along the bottom flange are shown in Figure 7. They are seen to be quite similar. The theoretical stresses shown are computed on the basis that the whole girder section resists lateral load. The fact that the measured stress closely approximates the theoretical stress shows that the girder acts very nearly as a unit under the action of the lateral load. At the center of the span it will be noted that the measured stresses on the top flange are a little larger than the theoretical, whereas on the bottom flange they are a little smaller. This is probably due to secondary torsional stresses at the center of the crane. Since the direction of twist reverses at the bridge center there can be no warping due to twist at this section. It is not likely that the stress variation is due to the top flange taking an appreciably greater proportion of the lateral load than the bottom flange, because the difference between the deflections of the top and bottom flanges (corrected for slip) is a smaller percentage than the measured variation in stresses. This is further substantiated by the fact that except at the center, the measured stress values lie very close to the theoretical values.

Figure 8 shows the variation in the average web stress along the girder. It should be remembered that the stresses in these tests are measured only on the outside of the girders since it is impossible to get inside the girder. Therefore the measured stresses represent the average stress of the members only when they are fairly straight in comparison with their thickness. This is substantially true for flange members, but not neces-

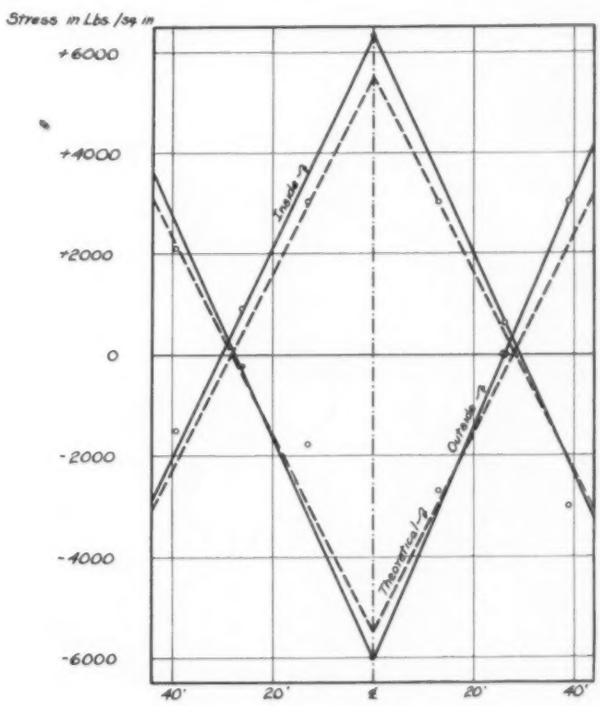
Figure 5



LATERAL LOAD TEST ~ CRANE #4683
STRESS DISTRIBUTION AT CENTER OF GIRDER

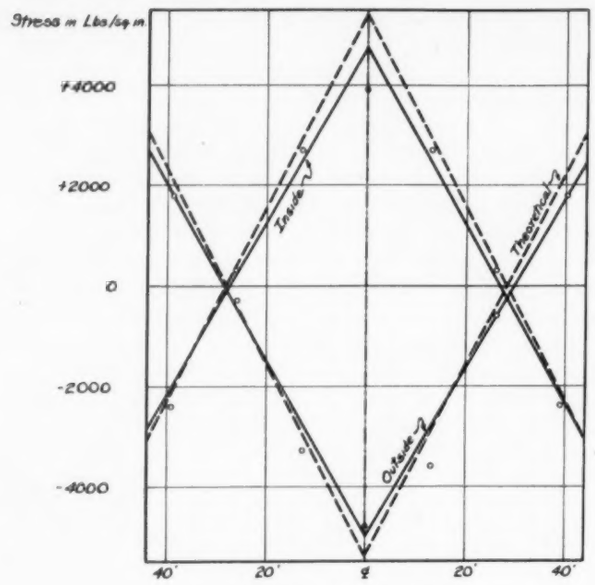
March 16, 1940

Figure 6



LATERAL LOAD TEST
CRANE #4683
TOP COVERPLATE STRESSES

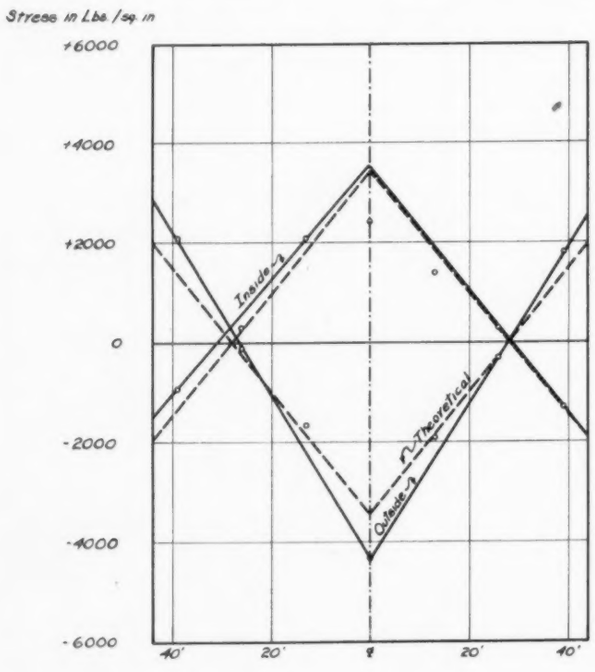
Mar. 14, 1940



LATERAL LOAD TEST
CRANE #4683
BOTTOM FLANGE STRESSES

March 16, 1940

Figure 7



LATERAL LOAD TEST
CRANE #4683
WEB STRESSES

Mar. 16, 1940

Figure 8

sarily true for web members; therefore the measured stress curves for the webs may not give the average stress, particularly when the web is thin.

In the stress curves for this crane the point of zero stress is seen to be at an average distance of 16 ft. from the ends of the girder. A point of contraflexure shows that some fixity is present. In this case the location of the point of zero stress corresponds to a partial fixity of 73 per cent. The statement 73 per cent partial fixity means that the moment at the end of the girder is 73 per cent of that which would be present if the girder were acting as a fully fixed beam.

The slip which occurred in the joint reduced the fixity. In addition, it is impossible to get 100 per cent fixity, since the elastic deflection due to the stress in the end carriage reduces the maximum possible end fixity to a value below 100 per cent. The fixity of the two girders varies due to the stiffening effect of the walkway. The 73 per cent fixity factor given above applies to both girders normally only when they have the same lateral moments of inertia. However, the stiffness factor of the end tie (moment of inertia divided by the length) is enough larger than that of the bridge girders so that there is no great difference between the points of contraflexure of the bridge girders. Therefore the 73 per cent fixity factor is a fair approximation. It should be remembered that no attempt is usually made to compute or allow for end fixity in design, in which case the end fixity is usually an added factor of safety. The theoretical stresses shown in the diagrams were computed using an end fixity of 73 per cent and a value of 4850 in.^{4} for the gross horizontal moment of inertia. The check is seen to be very close.

As an additional check the deflection was computed using 4850 in.^{4} for the moment of inertia of the idler girder and 17,540 in.^{4} for the moment of inertia of the motor girder. The difference between the two values is due to the walkway which was on the motor girder when the crane was tested. The computed deflection of the idler girder is 0.972 in. and that of the motor girder is 0.268 in. The average deflection is 0.620 in. The measured average lateral deflection as shown in Figure 4 is 0.605 in., and provides a good check. As stated before, the partial fixity factor of 73 per cent could be used for both girders only because they had approximately the same point of contraflexure. The influence of the walkway upon the lateral deflection of the motor girder should be noted as it reduces the deflection of this girder to 0.275 times the value it would have had, were the walkway not there.

No stress readings were taken at the 4400 lb. load but the deflections were measured. The deflection of 0.262 in. corresponds to a partial fixity of 60 per cent. The fixity at the higher load is greater than at the lower load. This is probably due to the fact that at the higher load, enough deformation had occurred to increase the number of bolts in bearing in the end tie, thus increasing the restraint.

It should be remembered that the design lateral load is about 1900 lb. However, this load in the test would have resulted in stresses too low to measure. The fixity will vary somewhat with the load. At very low loads the friction of the bolts carries the moment. As the load increases, there is a slip which reduces the fixity but as the bolts start to bear, the fixity increases. This is

shown by the increase in end fixity between the 4400 lb. and 12,600 lb. loads. The section acting to resist the lateral load does not vary with the load.

The theoretical deflections were computed by the following formula:

$$d = \frac{wl^3}{48EI} - \frac{Ml^2}{8EI}$$

M is the end moment as found from the measured stresses, w is the center load, l is the span length, I is the moment of inertia, and E is the modulus of elasticity.

The twist was computed by the formula:

$$\Theta = \frac{M_t l}{GK}$$

where Θ is the total angle of twist, G is the shear modulus, and K is the torsional constant. The torsion constant as computed from Bredt's formula is equal to the relation:

$$K = \frac{4A^2}{\int \frac{ds}{t}}$$

which, for a box section gives:

$$K = \frac{2b^2d^2t\,t_1}{bt_1 + dt}$$

where b and d are the width and depth respectively, t_1 equals the web thickness, and t is the average of the top and bottom flange thicknesses. In this crane, the torsional stiffness factors of both girders are practically alike, since, although the walkway changes the lateral moment of inertia markedly, it changes the torsion constant less than 1 per cent.

The results of the tests on riveted crane No. 4683 can be summarized as follows:

1. The whole section of the girder resisted the lateral moment.

2. Partial end fixity due to slip was present between the bridge girder and end tie. Its value varied from 60 to 73 per cent as the load varied.

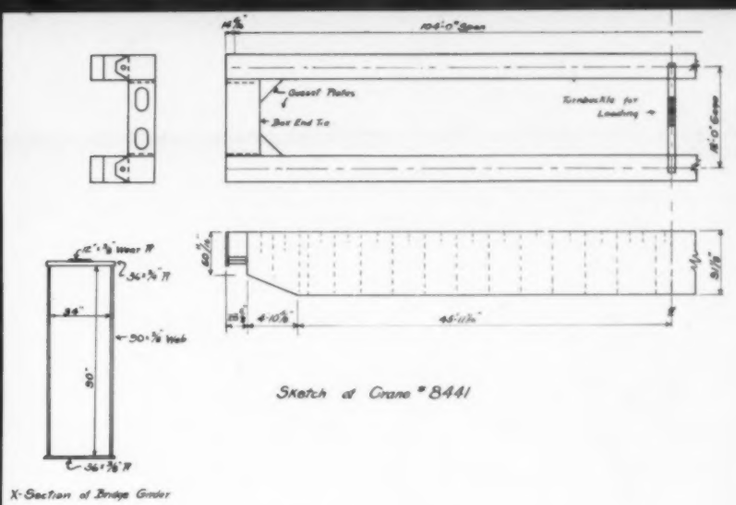
3. The twist of this riveted girder could be computed by Bredt's theory showing that there was no appreciable rivet slip to reduce the torsional rigidity. It should be noticed that the shearing stress due to torsion was only about 300 lb. per sq. in.

4. Secondary direct stresses due to torsion were present at the center of the girder. They were of small magnitude.

5. The walkway along the motor girder reduced the lateral deflection of this girder 75 per cent as compared with the deflection of the idler girder. Its effectiveness may be due to the fact that it was fastened to the web for its whole length and thus acted integrally as a stiffener on the girder.

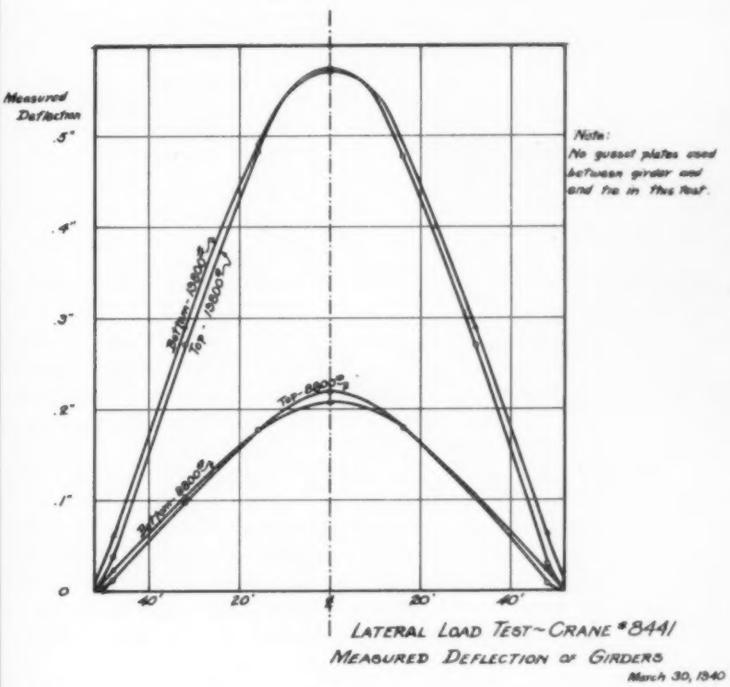
Crane No. 8441—The tests on welded crane No. 8441 were similar to those on No. 4683 and the discussion in this section will be devoted primarily to those points which differ from crane No. 4683.

Crane No. 8441 (see Figure 9) had no walkways on it during the tests. The girders were first tested with no gusset plates between the end tie and bridge girder and the lateral deflection curves are shown in Figure 10. Similar curves for the tests in which gusset plates were used are shown in Figure 11. The deflection of the girder with the gusset plates is less than the girder



Sketch of Crane #8441

Figure 9

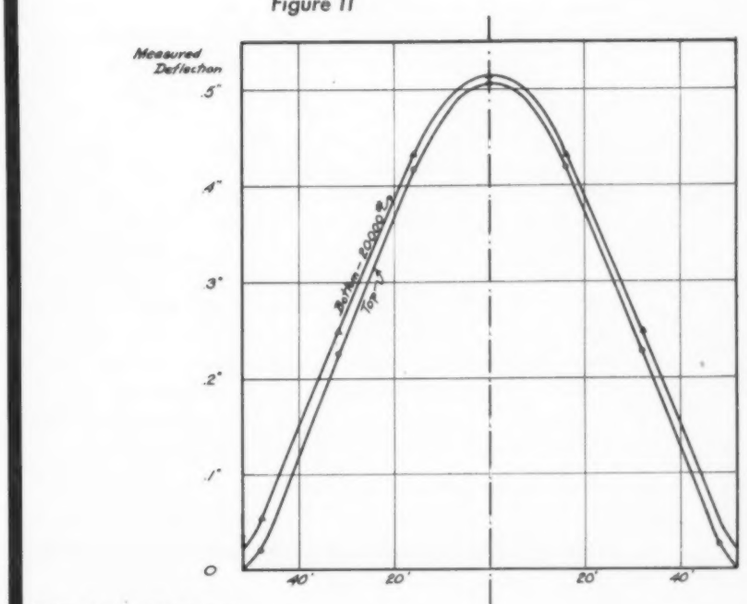


LATERAL LOAD TEST ~ CRANE #8441
MEASURED DEFLECTION OF GIRDERS
March 30, 1940

Figure 10

▲

Figure 11



Note: Gusset Plates used between girder and end tie in this test.

LATERAL LOAD TEST ~ CRANE #8441
MEASURED DEFLECTION OF GIRDERS
April 1, 1940

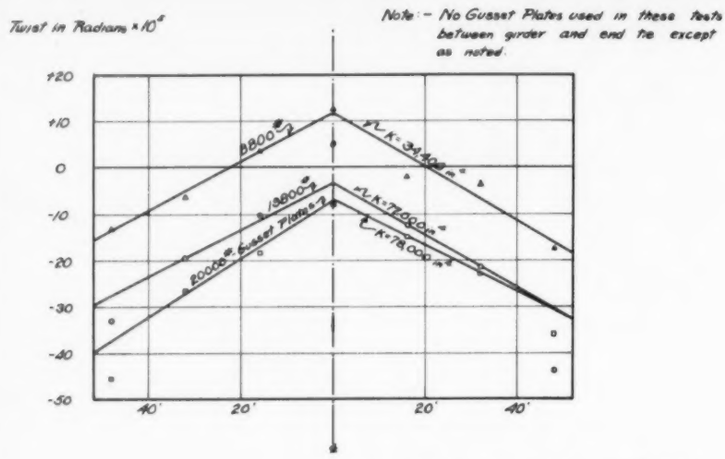


Figure 12

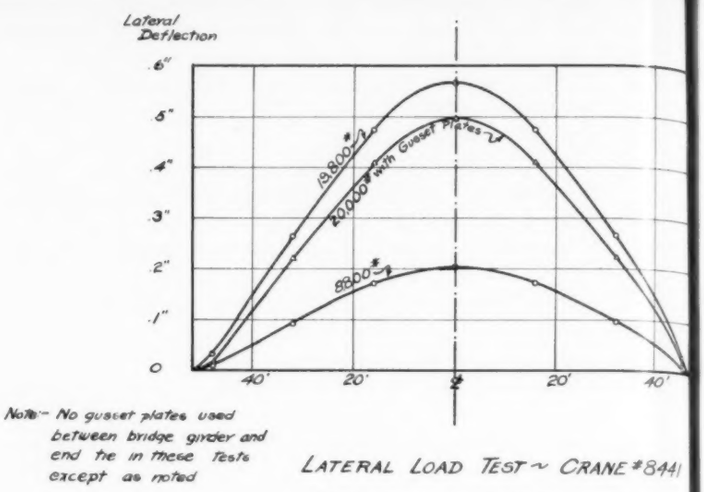


Figure 13

without the gusset plates due to the fixing effect of these plates. The fact that the deflection is greater on the bottom than on the top flange is due to slip in the end connection. The end tie was more effective in preventing slip at the top than at the bottom because the bottom of the end tie was located close to the center of rotation of the girder so that a small slip at this point would cause a large deflection on the bottom of the girder. Secondly, the bottom connection did not have as many bolts. That this slip occurred could be seen easily since the bolt holes were reamed prior to the test and they were out of line when the test was over. The resultant twist due to the slip is therefore somewhat contrary to what one might expect.

The measured twist of the girders is shown in Figure 12. The negative values are caused by the end slip. Since the measured twist is so small, the points are somewhat scattered and it is difficult to plot the correct curves. The difference between the K values of the 8800 lb. and 19,800 lb. loads is probably due to this fact. The measured values of K were found by dividing the torque by the product of the shear modulus and slope of the twist curve. Bredt's theory would give a K value of 61,000 in.^{4} However, since the twist angle is extremely small, it is believed that the difference from the theoretical is here a matter of precision. Since the twist was found by taking the difference of two sets of readings (the lateral deflection of the top and bottom flanges), whose values were very close to each other, a very small error in the measured values would give a rather large percentage error in the difference.

The lateral deflections shown in Figures 10 and 11 corrected for initial slip and twist are shown in Figure 13.

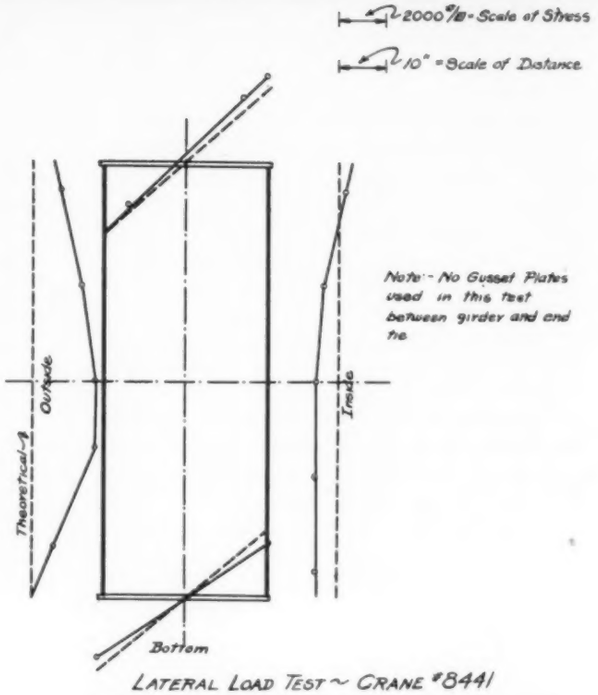
The stresses were measured on the front girder. Figure 14 shows the stress distribution at the center section of the girder and the plotted points show where the gauge lines were located. The stress distribution in the web should be noted. On the outside or compression side, very little stress is taken in the mid portion of the web. This is due either to local bending or incipient buckling of the web or perhaps to a combination of these two effects. Under lateral loads, the web acts as the compression flange of a beam and the h/t ratio of 240 for the clear depth of the web is so high that the stress is not apt to be uniform over the web cross section.

The test was repeated a number of times, similar readings were taken and the results were very inconsistent, showing that the webs were unstable at this stress. This indicates incipient buckling, since real buckling would overstress the flanges. The local bending effect was due to the fact that the welding of the diaphragms to the thin web plates caused small waves or buckles. Thus the stress caused by the test load produces bending due to eccentricity of the plate. Therefore the measured stress on one side of the web does not necessarily indicate the average stress through the thickness of the web.

Figure 15 shows the stress distribution along the top flange for the test in which no gusset plates were used. Figure 16 gives the stress distribution in the web. The irregular behavior of the web is clearly shown.

Figure 17 shows the stress distribution along the top flange when gusset plates were used. The stresses are seen to be less than those in Figure 15 which represents the same test without the gusset plates.

The stiffening effect of these gusset plates also shows up in the location of zero stress. For Figure 15 the point of zero stress averages 16¾ ft. from the end of the girder which corresponds to a fixity factor of 67 per cent. For the girder with gusset plates, the point of zero stress was 18½ ft. from the end and this corresponds to a partial fixity of 74 per cent.

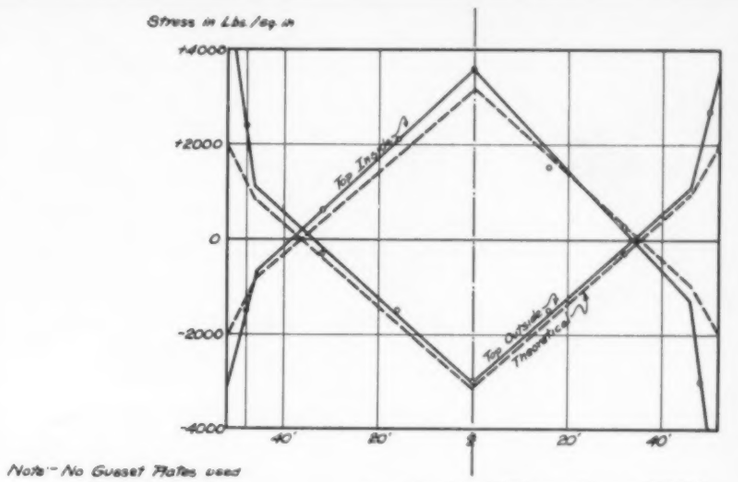

2000 #/in - Scale of Stress
10" = Scale of Distance
Note:- No Gusset Plates used in this test between girder and end tie
Outside
Inside
Theoretical
Bottom

LATERAL LOAD TEST ~ CRANE #8441

STRESS DISTRIBUTION AT CENTER OF GIRDER

Mar 30, 1940

Figure 14



LATERAL LOAD TEST ~ CRANE #8441

TOP COVERPLATE STRESSES

Mar 30, 1940

Figure 15

The span to use in computations is a questionable point but since the full span is normally used in design, the full span of 104 ft. was used in computing the above fixity factors. However, since the moment of inertia of the girder varied at the end, the point of contraflexure for the fully fixed beam was 25 ft. from the end instead of the 26 ft. which would have been the case if the moment of inertia were constant.

It is impossible to get 100 per cent fixity since the deflection in the end tie allows the end of the bridge girder to give. In crane No. 8441, the maximum possible end fixity is reduced to 84 per cent due to frame action. Thus the efficiency of the joint when gusset plates were used was 0.74/0.84 or 88 per cent, and 0.67/0.88 or 76 per cent when no gusset plates were used.

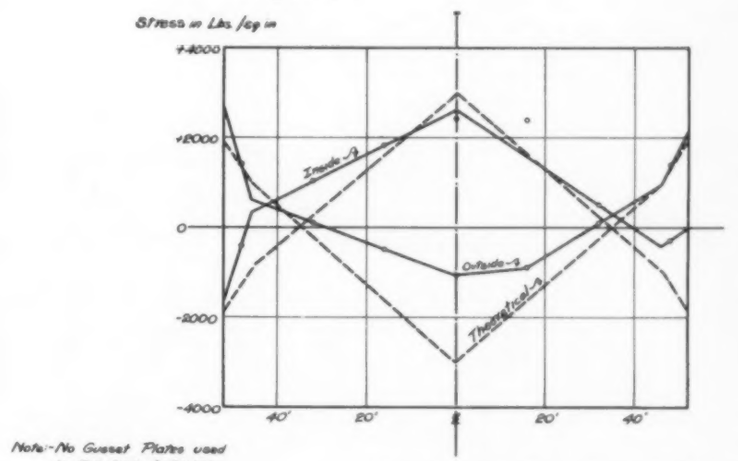
The theoretical stresses shown in the diagrams were computed using the end fixities found from the points of contraflexure and a value of 24,300 in.^{4} for the gross horizontal moment of inertia. The check is seen to be very close. The sharp break plotted in the stress lines is based on a consideration of the change in depth of the girder section.

As an additional check, the lateral deflection was computed using 24,300 in.^{4} for the moment of inertia and the fixity factors found from the stress curves. The theoretical deflection for the first test without the gusset plates is 0.572 in. and in Figure 13 the measured deflection is 0.564 in. For the test with the gusset plates, the computed deflection was 0.516 in. which compares with a measured value of 0.500. Thus the use of the whole girder to resist the lateral forces, and the values of the fixity factors is justified by the check between the observed and computed deflections.

The stresses at the 8800 lb. load were too small to measure with precision, but the lateral deflections were measured. The center deflection of 0.205 in. measured at this load corresponds to a fixity factor of 80 per cent and a joint efficiency of 91 per cent.

The design lateral load was about 14,200 lb. The

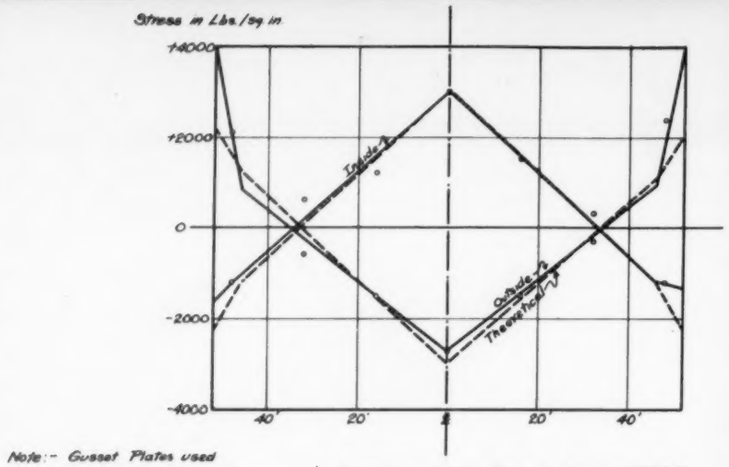
Figure 16



LATERAL LOAD TEST ~ CRANE #8441

AVERAGE WEB STRESSES

Mar 30, 1940



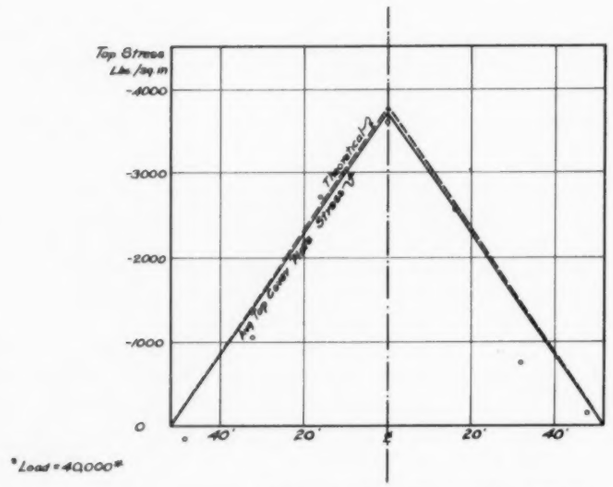
Note:- Gusset Plates used in this test between girder and end tie

LATERAL LOAD TEST ~ CRANE #8441

TOP COVERPLATE STRESSES

April 1, 1940

Figure 17



*Load = 40,000#

VERTICAL LOAD TEST~CRANE #8441

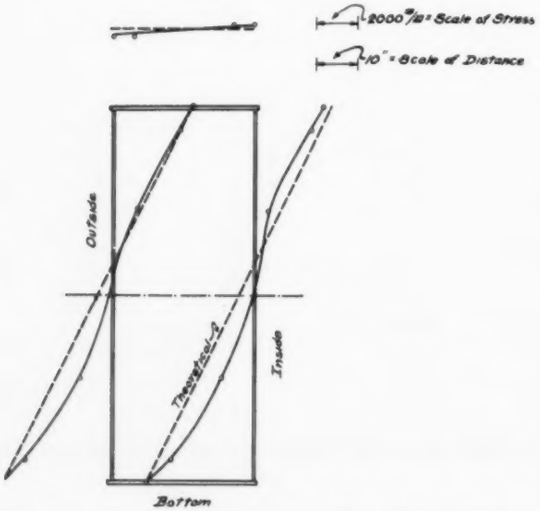
TOP COVERPLATE STRESSES

April 1, 1940

Figure 18

▲

Figure 19



Load = 40,000#

VERTICAL LOAD TEST ~ CRANE #8441

STRESS DISTRIBUTION AT ℄ OF GIRDER

Apr. 1, 1940

tests were therefore made at about 35 per cent overload and the stresses and deflection should be in line with what will occur when the crane is in service except as these factors are modified by the walkways, etc.

The end moments in the bridge girders caused corresponding end moments in the end ties. The stresses in the center of the end tie were found to be 3000 lb. per sq. in. at the corners of the box, which corresponds with a value of 3100 lb. per sq. in. computed from the fixed end moment. The test showed that the large manholes in the sides of the box end tie prevented the portion of the side plates in line with the manholes from carrying any stress. This was taken into account in the above computation.

The torsional moment had little effect on the end tie in comparison with the fixed end moment, since the lateral load had a small eccentricity with respect to the end tie. In this test, the twisting moment was less than 5 per cent the lateral moment.

Crane No. 8441 was also tested under vertical load. The girders were supported at the center with the ends cantilevered and the initial stress readings were taken. Then both ends were jacked up until the girders were raised off the center support and the strains measured again. This loading gave approximately the same bending moment as a load equal to the weight of the girder applied at the center of the girder.

The top coverplate stresses for this loading are shown in Figure 18. They check well with the ordinary beam theory as should be expected. Figure 19 shows the stress distribution at the center section of the girder under the action of vertical loads. The local bending in the thin web is shown by the refusal of the web to carry its full share of stress at the center line of the web while it does carry most of its share close to the flanges of the girder. This causes the curvilinear stress distribution in the webs.

The results of the tests on crane No. 8441 may be summarized as follows:

1. The whole section of the box girder resisted the lateral moment.

2. The end fixity for the girder with the gusset plates was 74 per cent for a 20,000 lb. load. The end fixity for the girder without the gusset plates was 67 per cent for a 19,800 lb. load and 80 per cent for an 8800 lb. load. This variation was due to slip in the end connection and deflection of the end tie.

4. The joint efficiency varied from 76 to 91 per cent.

5. Under vertical load, the box girder behaves as one would predict from the ordinary beam theory.

6. An h/t ratio of 240 for the clear depth of the web seems to be about the upper limit of this ratio or even somewhat above if the webs are to act as is usually assumed in design.

Crane No. 5886—The third crane tested was a fish-belly type and is sketched in Figure 20. This crane was tested with the walkway on each girder.

Figure 21 shows the lateral deflection curves of the flanges of this crane. In this case, the end deflection on the bottom is not zero and the slip caused the bottom to move outwards.

The twist of these girders is shown in Figure 22. It was rather surprising to note that the twist as measured in radians, was practically constant along the girder. However, these curves include slip. If the bottom

deflection curves of Figure 21 are raised so that the end deflections are zero, the effect of slip will be eliminated. This correction results in the corrected twist curves of Figure 22. For the 20,000 lb. load the center twist as measured is 0.00118 radians and the twist computed from equation (13) in the appendix gives a theoretical value of 0.00119 radians. Similarly for the 9300 lb. load, the measured value is 0.00058 radians and the computed value is 0.00056 radians. Equation (13) is based on Bredt's value for the torsion constant. The corrected twist curves are not parallel to the original twist curves due to the variable depth of the section.

The lateral deflection of Figure 21 corrected for slip is shown in Figure 23.

Figure 24 shows the stress distribution at the center section of the girder with the plotted points indicating the location of the gauges. It should be noted that the web stresses here are much closer to the theoretical than those of crane No. 8441. The web height-thickness ratio was the same in both cases, but crane No. 5886 had a straighter web and in addition was not stressed as highly as No. 8441.

Figures 25, 26, and 27 show the measured stresses along the flanges and webs of the girders. The theoretical stress distribution is not a straight line since the cross section of the girder varies. In Figure 25, the center stress was greater than the theoretical, and in Figure 27 it was less than the theoretical, whereas the rest of the girder checked fairly well. This is probably the same effect noticed in crane No. 4683 and is due to secondary torsional stresses.

The point of contraflexure in the stress curves are at an average distance of 22 ft. from the ends of the girder, which corresponds to a fixity factor of 84 per cent.

The lateral deflection was computed using the equations in the appendix and found to be $2.10 - 1.21 = 0.89$ in. as compared with a measured value of 0.90 in. The value of 2.10 in. is the deflection which would take place if there were no end fixity, and the 1.21 in. value is the deflection due to the end moment and acts in the minus direction. The difference is the actual deflection. The importance of end fixity and of keeping the end connection tight is apparent if a laterally stiff crane is desired.

The walkways on this crane seemed to have little effect on the deflection or stresses, as the above computations were made disregarding the walkways. Strain measurements on the walkway also showed no stress. The reason for the greater effectiveness of the walkway of crane No. 4683 as compared with that of crane No. 5886 is probably due to differences in connection details. A large portion of the walkway on crane No. 5886 hung below the girder and could have no effect. Secondly, the bolts of the walkway may have been looser than in the case of crane No. 4683.

The results of the tests on crane No. 5886 may be summarized as follows:

1. The whole section of the box girder resisted the lateral moment.

2. Partial end fixity was present which reduced the lateral deflection over 50 per cent. The fixity factor was 84 per cent.

3. Secondary torsion stresses were present which

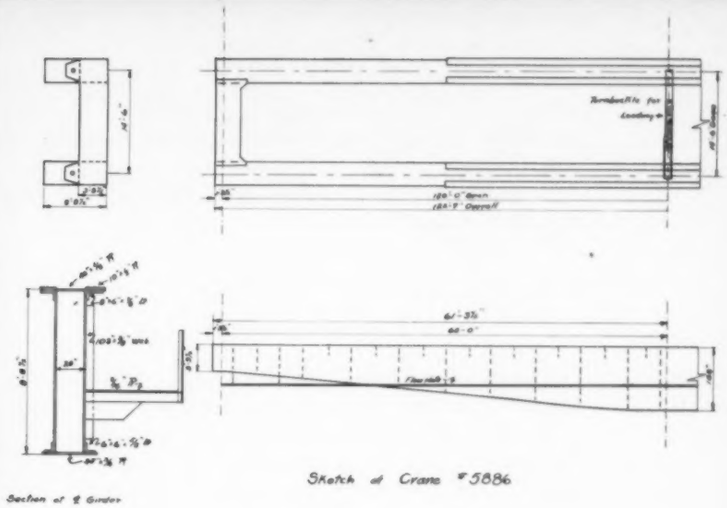


Figure 20

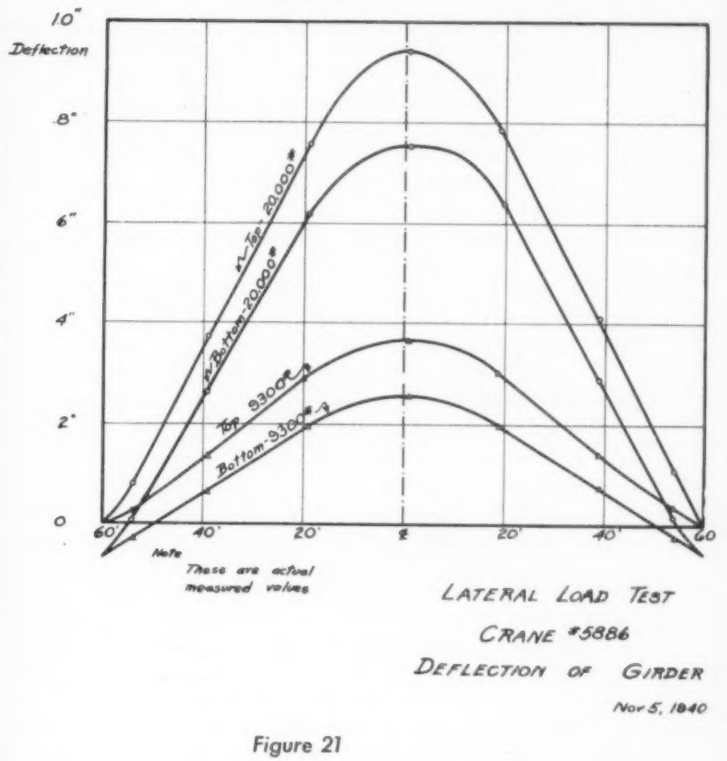
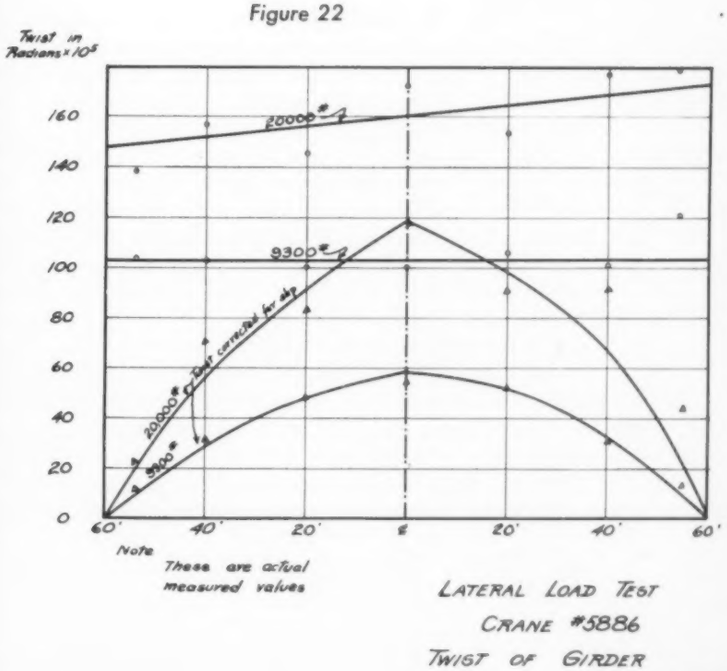


Figure 21

▲

Figure 22



increased the top flange stresses and reduced the bottom flange stresses.

4. The walkway carried no stress and had little effect on the behavior of the girder.

DISCUSSION OF RESULTS

The main purpose of the tests was to determine what portion of a box crane girder resists lateral loads. Since three very different types of girders were tested with the same result, one may conclude that the whole section of a box girder works to resist lateral loads. Therefore it seems logical that girders should be designed on this basis, in order to correlate design procedure with actual conditions. This should result in economy, since the usual method of assuming that the top flange takes the lateral load results in computed lateral stresses which are often three to four times more than those actually present. It should also result in a safer section, since the stresses in the tension flange due to lateral load, which are normally neglected, will be allowed for in the design.

The torsional direct stresses in the usual section will be quite small and should have little effect in the normally proportioned girder. They occur only at the point where the girder is loaded and become negligible within a few feet from this section.

End fixity was present in all the tests. This fixity is normally neglected in design. This neglect is probably justifiable since end fixity will vary greatly due to detailing and design. Secondly, the play in the end connections which appears as the crane is in service would reduce any fixity present to a small degree. Every crane maintenance man is familiar with the occurrence of loose bolts and rivets in the end tie connections. It must be remembered that the tests were made on newly fabricated cranes whose joints would be tighter than normal. Thus unless the joint is designed so that no slip can take place, it should be neglected in computations. Any fixity present will be an additional safety factor and help in stiffening the girder. However, the end tie should be designed to resist the end moment since neglect of this moment may result in overstressing the end tie.

The walkway worked with the girder in one test and had no effect in another. Since the walkway also loosens in service, it is not believed that it can be taken into account in design, unless special means are taken to make it act with the girder.

With the exception of one twist curve, the elastic twist could be computed fairly closely by Bredt's theory, and it is thought that the observed variations are due primarily to precision. However, the slip occurring in the end joints may cause a twist which is large in comparison with the elastic twist. This slip would increase as the crane is used in service.

Only one test was made under vertical load, and the results checked the ordinary beam theory. There is no reason to believe that the results should be otherwise.

Two of the cranes, No. 8441 and No. 5886, had relatively thin webs, the ratio of the clear depth to the plate thickness being about 240 in each case. One of these webs which had some small initial buckles, had rather

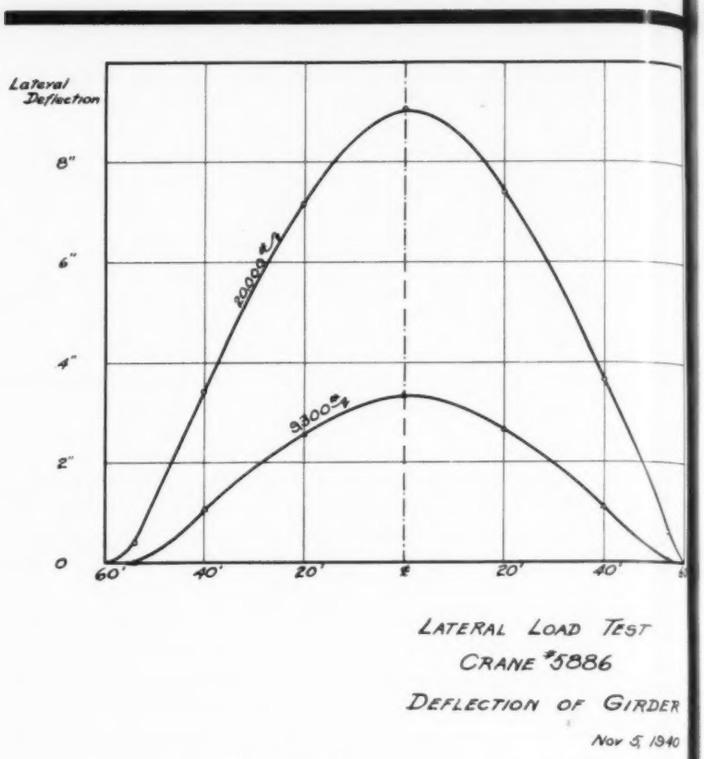
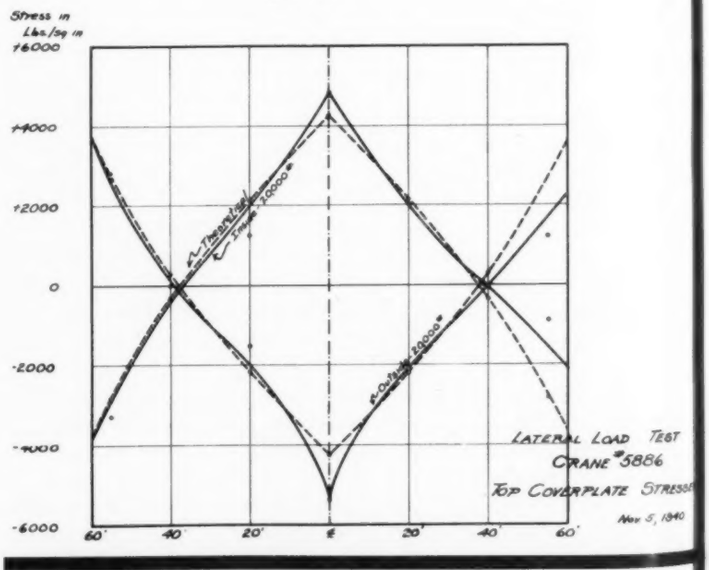


Figure 23

large bending stresses. The riveted web also had some bending stresses, but they were not as large as those in the welded web. However, before any comparison is made between the two, one should note that there was a diaphragm close to the center section of the riveted girder, whereas the center section of the welded girder was half-way between the diaphragms. This vertical

Figure 25



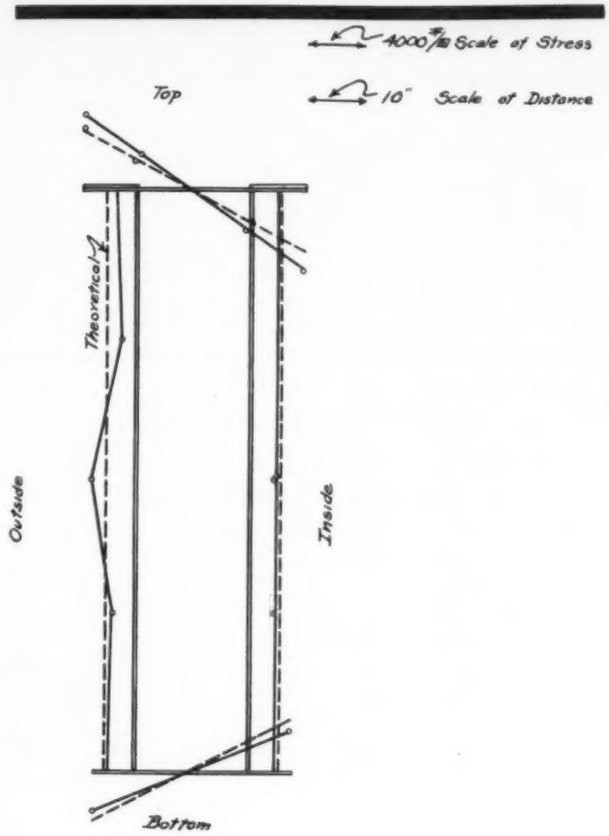


Figure 24

diaphragm would have some effect in reducing the bending stresses at the center section of the riveted girder. However, if thinner webs are to be used, it might be advisable to add longitudinal stiffeners. The above discussion applies primarily to the girder under the action of the lateral load only. In service the crane would have a vertical load in addition to the horizontal load and the combination of the two loads would usually result in a lower allowable web height-thickness ratio than if the crane were under the action of either load alone.

CONCLUSIONS

1. The results of these tests show that the whole girder section acts to resist lateral loads and should be so considered in design.

2. The stresses of a crane under vertical load can be computed by the ordinary beam theory.

3. The elastic twist of a girder can be computed approximately, by Bredt's theory. In addition to the elastic twist there is quite a bit of twist due to slip.

APPENDIX

The following formulæ were developed for fish-belly girders from the fundamental differential equation

$$E\frac{d^2y}{dx^2} = -\frac{M}{I} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (1)$$

where y is the deflection of the girder at any point x measured from the end of the girder.

Since I is a variable in this equation as well as M, some assumption must be made as to the variation of I. Two sets of formulæ were derived, one in which it was assumed that I varied linearly along the girder, and second, it was assumed that I varied parabolically along the length of the girder. The first approximation is most applicable to computing the lateral moment of inertia, and the second in computing the vertical mo-

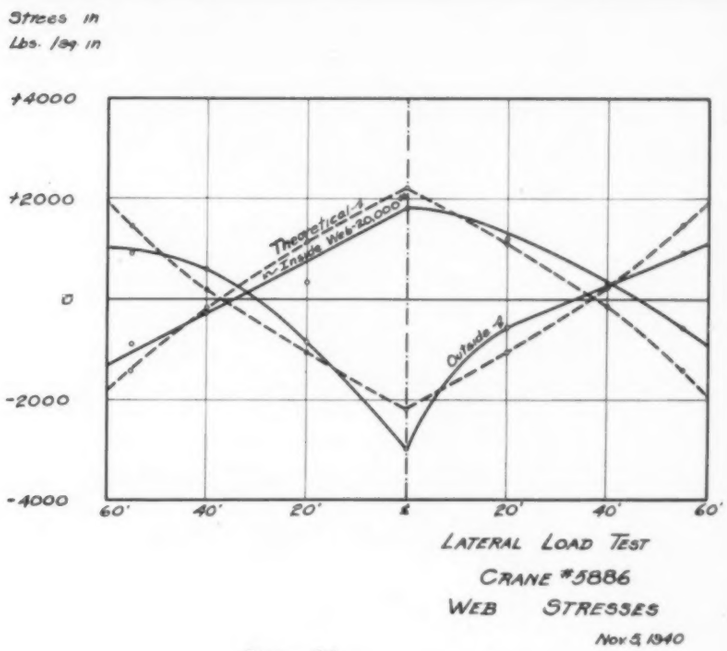
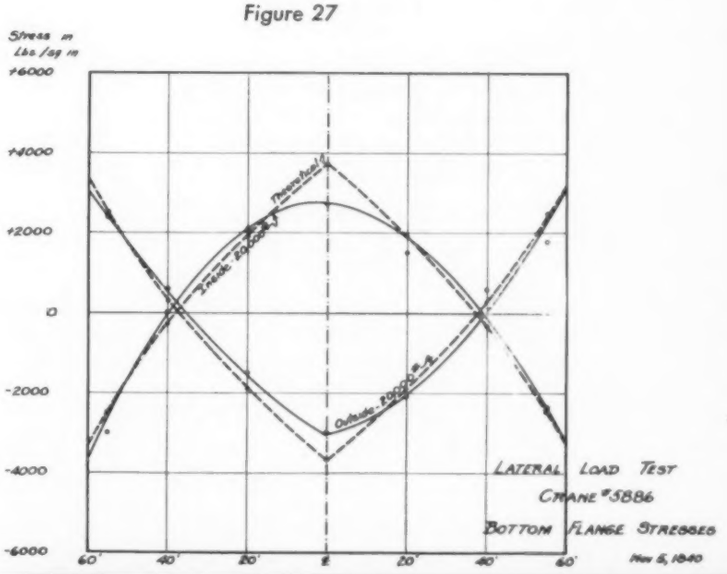


Figure 26

Figure 27



ment of inertia. For average girders they should give results which are fairly close.

The computations for deriving these formulæ are quite laborious and only the results will be given.

For a girder in which the I varies parabolically, the deflection at any point x due to a load at the center of the girder is

$$d=\frac{-Pl^2}{16EI_3}\left[x\log\left(\frac{I_1l^2}{I_3^4}+x^2\right)-2x+2\sqrt{\frac{I_1l^2}{4I_3}}\left(\tan^{-1}\frac{x}{\sqrt{\frac{I_1l^2}{4I_3}}}\right)-x\log\left(\frac{I_1l^2}{I_3^4}+\frac{l^2}{4}\right)\right] \quad \ldots\ldots(2)$$

and for the center deflection, this reduces to

$$d_c=\frac{-Pl^3}{16EI_3}\left(-1+\sqrt{\frac{1}{\frac{I_2}{I_1}-1}}\tan^{-1}\sqrt{\frac{I_2}{I_1}-1}\right) \quad \ldots\ldots(3)$$

P stands for the center load in pounds, l is the span in inches, I_1 is the gross moment of inertia at the end, I_2 is the gross moment of inertia at the center, I_3 equals I_2-I_1, and E is the modulus of elasticity. The above formulæ give the deflection for a simple beam.

The corresponding formulæ for linear variation of I are:

$$d=\frac{-Pl}{8EI_3^2}\left[I_3x(x-l)+I_1xl\log\left(\frac{I_1l+I_3l}{I_1l+2I_3x}\right)-\frac{I_1^2l^2}{2I_3}\log\left(1+\frac{2I_3x}{I_1l}\right)+I_1xl\right] \quad \ldots\ldots(4)$$

and for the center deflection

$$d_c=\frac{-Pl^3}{16EI_3}\left[-\frac{1}{2}+\frac{I_1}{I_3}-\left(\frac{I_1}{I_3}\right)^2\log\left(1+\frac{I_3}{I_1}\right)\right] \quad \ldots\ldots(5)$$

The deflections due to the end moment are given by the following formulæ where M is the end moment.

For parabolic variation of I

$$d=\frac{-Ml^2}{4EI_3}\left[\frac{2x}{l}\sqrt{\frac{I_3}{I_1}}\tan^{-1}\frac{2x}{l}\sqrt{\frac{I_3}{I_1}}-\frac{2x}{l}\sqrt{\frac{I_3}{I_1}}\tan^{-1}\sqrt{\frac{I_3}{I_1}}-\frac{1}{2}\log\left(1+\frac{4I_3x^2}{I_1l^2}\right)\right] \quad \ldots\ldots(6)$$

and for the center deflection, this reduces to

$$d_c=\left(\frac{Ml^2}{8EI_3}\right)\log\left(1+\frac{I_3}{I_1}\right) \quad \ldots\ldots(7)$$

For linear variation of I

$$d=-\frac{Ml}{2EI_3}\left[x\log\left(\frac{I_1l+2I_3x}{I_1l+I_3l}\right)-x+\frac{I_1l}{2I_3}\log\left(\frac{I_1l+2I_3x}{I_1l}\right)\right] \quad \ldots\ldots(8)$$

and for the center deflection, this reduces to

$$d_c=\frac{-Ml^2}{4EI_3}\left[-1+\frac{I_1}{I_3}\log\left(1+\frac{I_3}{I_1}\right)\right] \quad \ldots\ldots(9)$$

The fixed end moment is given by the following formula for linear variation of I. There would be no occasion to compute it in the vertical direction.

$$M=\frac{Pl}{4}\sqrt{\frac{I_1}{I_3}}\frac{\left[-1+\frac{I_1}{I_3}\log\left(1+\frac{I_3}{I_1}\right)\right]}{\tan^{-1}\sqrt{\frac{I_3}{I_1}}} \quad \ldots\ldots(10)$$

The actual fixed end moment would vary from 60 to 90 per cent of the above value depending on the end connection.

In the above equations the units must be consistent, and it is advisable to use pounds and inches. The angles are expressed in radians and the logarithms are natural logarithms to the base e and not to the base 10.

These equations require a good deal of computation for their solution, except for the deflections at the center where they are considerably simplified. The effect of varying the moment of inertia is to increase the deflection over what would take place if the whole girder section were equal to that at the center. For a number of girders which have been checked by the writer, the increase in deflection has varied from 10 to 30 per cent. Therefore for approximate computations, the following formula is suggested for the center deflection of fish-belly girders.

$$d_c=K\left[\frac{Pl^3}{48EI_c}-\frac{Ml^2}{8EI_c}\right] \quad \ldots\ldots(11)$$

I_c is the moment of inertia at the center, M is the fixed end moment which will vary from 60 to 85 per cent of the fully fixed beam depending on the connection, and K is a constant to take into account the fish-belly effect. K varies from 1.10 to 1.30, the smaller value applying to relatively flat fish-belly girders and the higher value to deep fish-belly girders. For a girder of constant section K equals 1.00.

In a fish-belly girder, the twisting moment due to a lateral load applied at the top flange is not constant but varies along the girder since the center of gravity of the girder approaches the top flange as one approaches the end.

The twist for this condition is given by the following formula, when the depth of the girder varies parabolically.

$$Y=\frac{Pl}{16b^2G}\left[\frac{b(t_2+t_1)}{2t_1t_2\sqrt{h_1h_3}}\tan^{-1}\left(\frac{2x}{l}\sqrt{\frac{h_3}{h_1}}\right)+\frac{2x}{lt_3}\right] \quad \ldots\ldots(12)$$

and the twist at the center

$$Y_c=\frac{Pl}{16b^2G}\left[\frac{b(t_2+t_1)}{2t_1t_2\sqrt{h_1h_3}}\tan^{-1}\sqrt{\frac{h_3}{h_1}}+\frac{1}{t_3}\right] \quad \ldots\ldots(13)$$

Y is the angular twist in radians, b is the distance between the web centers, G is the shear modulus, t_1 and t_2 are the cover plate thicknesses, t_3 is the web thickness, h_1 is the depth of web at the end, h_2 is the center web depth, and h_3 is the difference between the center web depth and end depth.

However, for approximate computations, we may use for the center twist

$$Y_c=\frac{Pl}{32b^2h_2G}\left[\frac{b}{t_1}+\frac{b}{t_2}+\frac{2h_2}{t_3}\right] \quad \ldots\ldots(14)$$

This formula is simply the formula for the uniform section, and its use is based upon the assumption that the resistance to twist and the applied moment at any section vary proportionately. For the crane tested, the computed value given by equation (14) is practically the same as that given by the more exact equation (13).

The twist caused by a uniform torque along this girder, such as the motor torque may be computed by the following formula.

$$Y=\frac{Ml}{4Gb^2}\left[\left(\frac{b}{t_1}+\frac{b}{t_2}\right)\frac{lx}{2h_1(h_1l^2+4h_3x^2)}+\left\{\left(\frac{b}{t_1}+\frac{b}{2}\right)\frac{1}{4h_1}+\frac{1}{t_3}\right\}\frac{1}{\sqrt{h_1h_3}}\tan^{-1}\frac{2x}{l}\sqrt{\frac{h_3}{h_1}}\right] \quad \ldots\ldots(15)$$

and the twist at the center is obtained by substituting $x = \frac{l}{2}$ in the above formula.

The above formulæ are given here as a matter of record since it was necessary to derive them in order to check the test results. Their application in practice would probably be laborious. As a matter of fact, a properly designed box girder is so stiff that there is little necessity for computing twist.

The formulæ which might have a practical application are those for lateral deflection since they are a measure of the stiffness of the crane. Also, an average crane often has a lateral deflection under the design load which is greater than the vertical deflection. Since this lateral deflection occurs in both directions, there can be no camber; and the resultant wear on shafting, bearings, etc., is more serious than that due to the vertical bending.

The above equations apply when the girders are loaded at the center, and this condition gives the maximum possible deflection very closely.

2. Lateral Load Tests on Truss I Beam Crane

▲ THIS REPORT summarizes the results of a lateral load test made on the bridge of a 175 long-ton trussed I-beam crane of 100 ft. 11 in. span. The stresses and deflections in the crane were measured for the lateral test load of 23,700 lb. The whole truss was found to resist the lateral load. Secondary stresses were present in the truss. The stresses in the crane were computed by three methods, and the computed results are compared with the measured results.

Since little is known about the behavior of trussed I-beam cranes, this test was made in order to compare actual behavior with design assumptions.

Crane No. 5879, which is a sixteen-wheel crane, was supported on the equalizer trucks during the test. The crane was tested after it had been completely assembled, but without the trolley.

The stresses were measured with a 10 in. Whittemore strain gauge which gives results accurate to about plus or minus 300 lb. per sq. in. They were measured on the girder, end ties, and on a number of the more highly stressed truss members. Since the crane was due for shipment, only a limited time was available for the test and it was impossible to take as many readings as would normally be desirable. Sketches showing some of the details of the girder and the truss members discussed in this report are given in Figures 1 and 2.

The lateral deflection of the girders was determined by means of a 0.001 in. Federal dial which measured the closure as the loading rig pulled the two girders together. The difference between the lateral deflections of the top and bottom flanges at the same section divided by the distance between the gauge points gave the twist of the girder at each section.

The load was applied by tightening a turnbuckle fastened to the top flanges of both girders at the center of the span. The load was measured by a calibrated spring which was placed in the system.

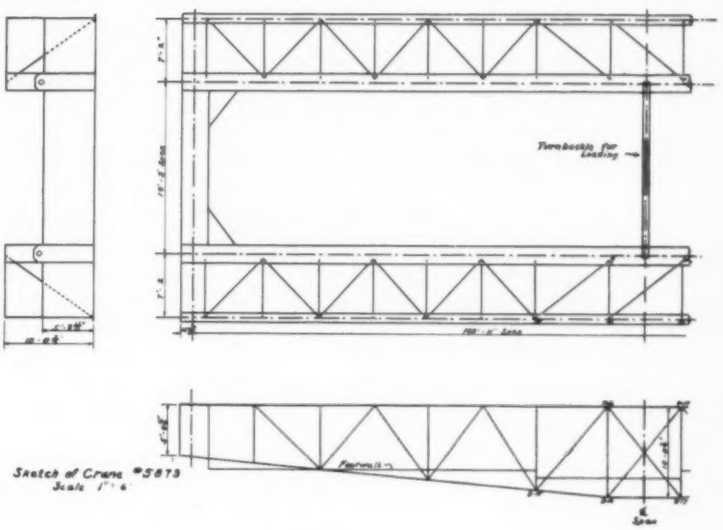
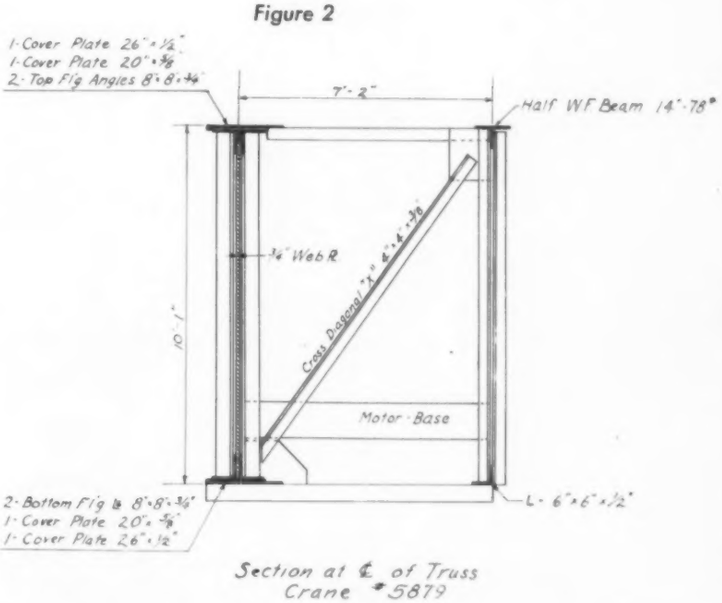


Figure 1

Figure 2


1-Cover Plate 26" x 1/2"
1-Cover Plate 20" x 3/8"
2-Top Flg Angles 8" x 8" x 3/4"
7'-2"
Half WF Beam 14"-78"
3/4" Web R
Cross Diagonal 7" x 4" x 3/8"
10'-1"
Motor Base
2-Bottom Flg ⊾ 8"x8"x3/4"
1-Cover Plate 20" x 3/8"
1-Cover Plate 26" x 1/2"
L - 6" x 6" x 1/2"

Section at ℄ of Truss
Crane #5879

TEST RESULTS

The girders were loaded with a lateral load of 23,700 lb. applied on the top flange. This was the maximum capacity of the loading rig. Since the capacity of the crane was 175 long tons or about 392,000 lb., the test

load was about equivalent to a lateral load on each truss of 6 per cent of the live load capacity of the crane.

Figure 3 shows the lateral deflection of both flanges of the girder under the test load. The center line mark denotes the center of the girder span. The deflections are given in inches and are the measured deflections divided by two since the gauge gave the combined deflections of both girders.

The measured twist of the girders is shown in Figure 4.

The measured stresses due to the lateral load are shown in Figures 5, 6, and 7. Figure 5 shows the stress distribution along the top flange of the I-beam girder. The gauge lines on which these stresses were measured were located on the top coverplate 10 in. from the centerline of the I-beam. This figure shows a large bending stress at the center of the girder. The lateral load was applied at the span center and since this was not a panel point location, the I-beam transferred the load to these points. If the top flange alone is assumed to transfer the load as a continuous beam to the panel points, the computed stress will be plus or minus 2700 lb. per sq. in. Since there is a resultant tension in the girder of about 600 lb. per sq. in. due to truss action, the combined stress would be +3300 lb. per sq. in. and −2100 lb. per sq. in. The measured values are +3900 and −2700 lb. per sq. in., which are somewhat higher than the computed.

Figure 6 shows the stress distribution along the bottom flange of the I-beam. The stresses were measured on the edges of the flanges. The local bending at the load point had little effect on the bottom flange. This shows that the top flange transferred most of the load to the panel point.

In Figure 7 the average stresses of the top and bottom flanges of the girder are shown. These average stresses are a measure of the load carried by the various members. The stresses in the bottom truss are quite appreciable, and would be completely neglected in normal design procedure.

DISCUSSION OF RESULTS

In this country, truss cranes have not been as extensively used as in Europe, even though they may result in lighter cranes on long spans. Some of the reasons for this have been economic, and some have been due to a lack of data as to the stiffness and stress distribution in such cranes. This test was made to determine the latter factors and to compare actual measurements with common design assumptions. Although conclusions drawn from a single test can not be used as a basis for general conclusions, they indicate a trend. Secondly, the measurements do show how this particular crane behaved under the test load. Previous data of this type are very scarce.

Crane No. 5879 is a space frame; it is highly indeterminate, and the stresses can be found only after long, tedious computations. In addition, one side of the crane is a complete I-beam girder and little is known as to how much of the girder takes part in resisting the loads. Finally, the problem is complicated by the effect of the walkway and any end fixity which may be present.

Table I presents a comparison between the measured stresses and the stresses computed by three different

TABLE I

Comparison of Measured and Computed Stresses

Bar	Measured stress	Measured stress	Measured stress	Area	Average measured stress	Measured load, lb.	Computed stress as beam	Computed load as two separate trusses	Computed load as space frame
Top truss									
7– 8	+3900	−2700	..	90.00	+ 600	+54,000	+ 420	+53,200	+49,000
7–15	+4500	+4800	..	3.84	+4700	+18,000	..	+10,700	+13,500
7–16	+1800	+ 900	..	2.48	+1350	+ 3,400	..	+ 8,500	+ 4,400
16–17	−2100	−3600	..	11.50	−2850	−33,000	−4300	−53,200	−49,000
Side truss									
T17–B16	+ 600	−1200	..	2.48	− 300	− 750	..	...	0
T16–B16	+ 900	+1800	+1200	2.86	+1300	+ 3,700	..	...	+ 2,000
T16–B15	+ 300	+ 300	+ 900	2.48	+ 500	+ 1,240	..	...	+ 1,900
Bottom truss									
7–16	0	..	..	3.61	0	0	..	0	0
7–15	+ 900	+ 600	..	2.86	+ 750	+ 2,100	..	+ 6,000	+ 3,100
7– 8	+1200	− 600	..	90.00	+ 300	+27,000	+ 420	+29,800	+24,000
16–17	−3900	..	..	5.75	−3900	−22,500	−4300	−29,800	−24,000
15–16	−3600	..	..	5.75	−3600	−20,700	−3600	−29,800	−24,000
Diagonal *X*	−1200	−1800	− 300	5.72	−1100	− 6,300	..	− 7,200	− 7,400

TABLE II

Comparison of Measured and Computed Deflections

Location	Measured, in.	Computed as beam, in.		Computed as two trusses, in.		Computed as space framework, in.	
		No fixity	Partial fixity	No fixity	Partial fixity	No fixity	Partial fixity
Top flange center.....	0.350	0.276	0.237	0.374	0.334	0.417	0.380
Bottom flange center...	0.240	0.276	0.237	0.340	0.300	0.367	0.325

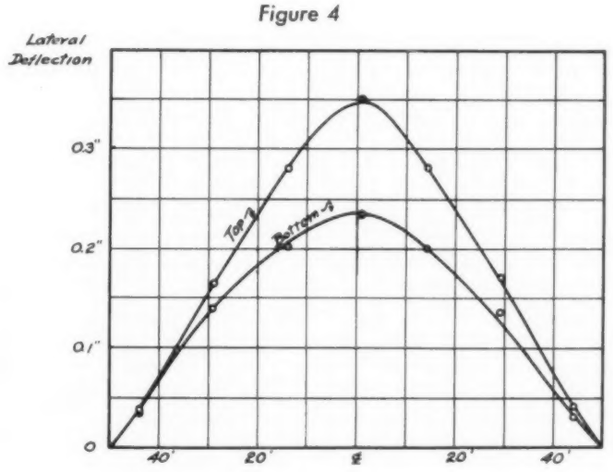
methods. Table II gives a similar comparison for them deflections. The most approximate computation method is to consider the truss as a beam. In crane No. 5879, the moment of inertia was computed by assuming the chords of the truss to be one flange of the beam and the whole I-beam to be the other flange. The computed I was 116,500 in.^{4}. The tables show that this results in a fair approximation for the stresses, and also for the deflections. However, deflections computed by this method will be low since the deflection of the structure due to the strain in the diagonals is neglected. Also this method of computation gives neither the stresses in the diagonals, nor the twist in the crane.

It should be noted in Table I that the stresses were taken on one, two, or three gauge lines on each member tested. These gauge lines were usually on the edges of the member and on the same section in every case. The tables show, however, that the stress at a section is not uniform. This variation is probably due to secondary moments in the truss. Trusses are normally considered to be pin-connected at the joints for computation purposes, whereas they are not usually so in practice. In this crane the members were welded to the gusset plates, and the chords were continuous through the truss. These secondary moments will cause bending stresses in the members, but should not appreciably affect the direct stresses. Secondly, the angles can be welded to the gusset plates on only one leg, which results in an eccentric application of the load on the member and consequent bending stresses.

Since there was some bending in the member, the measured stresses may not necessarily give the average stresses and this fact may account partly for some of the discrepancy between the computed and measured results.

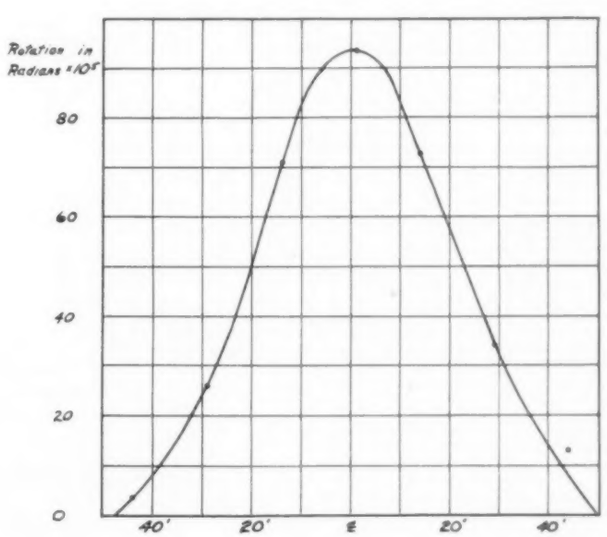
In practice, a crane of the type tested is often de-

Figure 3



LATERAL LOAD TEST
CRANE #5879
LATERAL DEFLECTION OF GIRDER
Nov 8, 1940

Figure 4



LATERAL LOAD TEST
CRANE #5879
ROTATION OF GIRDER
Nov 8, 1940

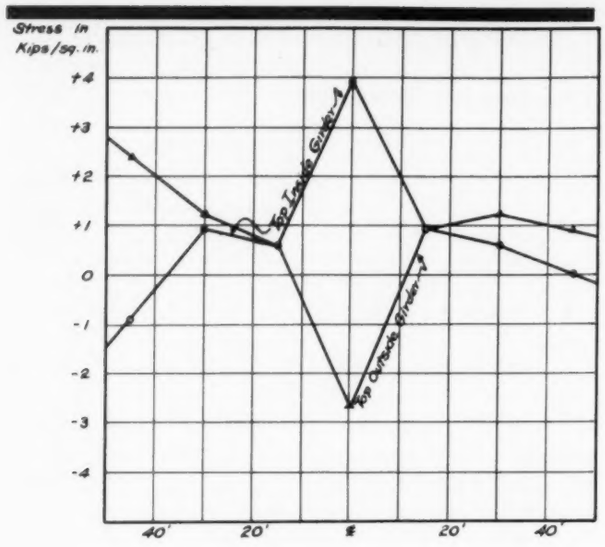


Figure 5

signed on the assumption that the top truss takes all the lateral load applied to the top flange. Some designers will split the load between the two trusses with the top truss taking the greater proportion. Figure 7 shows that the bottom truss is highly stressed due to a load on the top flange. Therefore one can not very well assume that the top chord takes all the load, and that the rest of the crane remains unstressed.

In the second method of computation, the load was split between the two trusses and each truss was analyzed as a separate truss. The method of equating deflections (*Structural Theory*—Sutherland and Bowman) was used to determine how much load was taken by each truss. In this method, the trusses are assumed to be pin-ended. It is also assumed that the center cross diagonals transfer the load from the top truss to the bottom. The diagonal is cut to obtain a statically determinate structure and is the member marked X in Figure 2. The deflections of each truss are then computed. The load taken by each truss is that which makes the relative deflection of the cut ends of the diagonal zero. In this crane, the computation showed that the top truss took 15,200 lb. and the bottom truss 8500 lb. of the applied 23,700 lb. lateral load. These results check approximately as the tables show. The method, however, neglects the effect of the side trusses and does not give the stresses in the side truss diagonals.

In the last column of the tables, the computed loads are given for the truss considered as a space frame. This computation method takes into account all the members. Two assumptions were made in the analysis. One was to split the area of the I-beam into two parts and to consider each part as one of the chords of the truss. Secondly it was assumed that only the two center cross diagonals X transferred load between the trusses. This second approximation is close since these diagonals are at the loaded panel points. When the cross-diagonals are cut, the various trusses are statically determinate for horizontal and vertical loads, and the stresses can be found by the method of deflections.

Since the structure is indeterminate, adding more area to a member does not necessarily reduce the stresses in that member. The result may be that the member will take more load. In this crane, the top chord of the outside truss had twice the area of the bottom chord. A lateral load of 100 lb. on the top chord results in a maximum force of 228 lb. in the top chord and a maximum force of 122 lb. in the bottom chord. On the other hand, if a lateral load of 100 lb. is applied on the

Figure 6

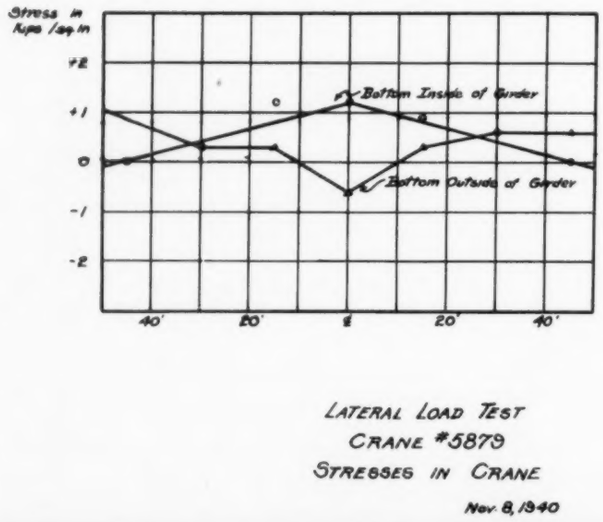
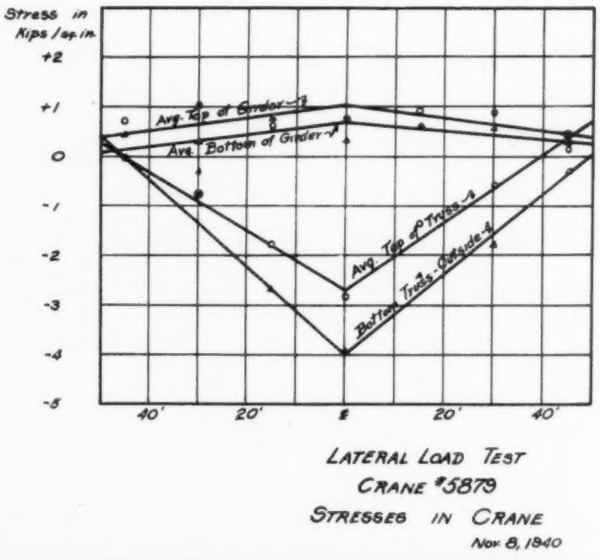


Figure 7



bottom chord, there is a computed maximum force of 141 lb. in the bottom chord and 209 lb. in the top chord. In other words, the lighter the loaded truss, the more load will be transferred to the adjacent trusses. In the case of the crane tested, the computations indicate that if the top and bottom truss were made equally strong, the maximum load in the chords would be less. The trusses would have more of a tendency to take an equal portion of the load, and there would be less twist in the crane.

Figure 7 shows a slight end moment in the trusses. Stress measurements in the end tie showed that this end moment was about 20 per cent of the fully fixed end moment. Since the truss was very stiff compared with the end tie, the maximum possible end fixity which could be developed if there were no slip in the joint, was 47 per cent.

Figure 3 gives the lateral deflection of the crane. If the lateral deflection shown here is compared with that of some box girder cranes previously tested ("Lateral Load Tests on Box Girder Mill Cranes"—Madsen), and adjusted to take into account the difference in capacity and span, it is seen that the truss crane is fully as stiff. Secondly, crane No. 5879 has so little initial end fixity, it will remain almost as stiff when the end ties loosen in service. This can be seen by looking at the values for the deflection for partial fixity and no fixity in Table II. The box girders previously tested had a much higher end fixity and loosening of the end tie connection would increase the lateral deflection markedly.

The measured deflections shown in Figure 3 are less than the computed deflections given in Table II. This is most likely due to the fact that the computations are made on the basis of pin-end connections and the welded joints would stiffen the truss. Secondly, the walkway may have had some stiffening effect, particularly on the lower chord.

If the twist of the crane shown in Figure 4 is compared with the twists of the box girders previously tested, it would appear that the box girders have more resistance to twisting.

The above comparisons, however, may not be overly significant since it is very difficult in a comparison of cranes of different capacities and spans to draw any definite conclusions.

CONCLUSIONS

The measurements made in this test showed that for the crane tested:

1. The whole truss resisted the lateral load as a space frame.
2. The whole I-beam girder acted as a part of the truss.
3. The stresses computed by the three different methods gave results which were a fair approximation of the measured loads. The solution of the truss as a space frame is the only method which could be used to find the stresses in all the members.
4. When the lateral load is not applied at a panel point, the top chord must be designed to transfer the applied load to the panel points.
5. Secondary stresses were present throughout the truss.

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3. Dynamic Tests on Mill Cranes

▲ THIS REPORT describes the results of impact and lateral load tests on three electric overhead traveling cranes at the Bethlehem plant of the Bethlehem Steel Corporation and on the crane in the Fritz Engineering Laboratory of Lehigh University. In these tests, the stresses in the girders due to acceleration, braking, and impact, were measured with scratch gauges. It was found that the maximum lateral stress was due to braking and was about 10 per cent of the vertical live and dead load for the cranes tested. The measured impact values were quite variable. Jerk impact was found to vary from 9 to 33 per cent of the live load. Secondly, when the crane was run over wedges to simulate bad runway joints, the impact was found to vary from 56 to 100 per cent of the live load.

These tests were made in order to determine whether specification requirements for dynamic stresses in mill cranes were in accordance with those actually present in the girders of such cranes.

The dynamic stresses in cranes are of two types. One type of dynamic stress is the lateral stress due to acceleration and braking of the crane. This stress may be very large when the crane is of long span and the girders are relatively narrow. The braking stresses are usually larger than the acceleration stresses. The ratio of the braking load to the vertical load on the braked wheels may approach the value of the coefficient of friction between the crane wheel and runway rail. This maximum value may be modified by the timing of the swing of the load on the crane. If the instant of maximum swing should occur with the maximum braking force, the ratio of the maximum lateral force to the vertical force may exceed the coefficient of friction. However, in the tests made, the swing of the pendulum lagged behind the braking force. Lateral forces may also be due to other causes than those above enumerated. One such case would be the use of cranes to spot railroad cars. However, even in such cases, the maximum ratio of lateral load to vertical load should not exceed the coefficient of friction, since at this point the crane would slip on the runway rail.

Impact stresses may be due to many causes. In mill practice running the crane over bad runway joints is probably the most important cause for such stress. Impact stress will also occur when a load is jerked off the ground.

Four cranes were tested in this program. One series

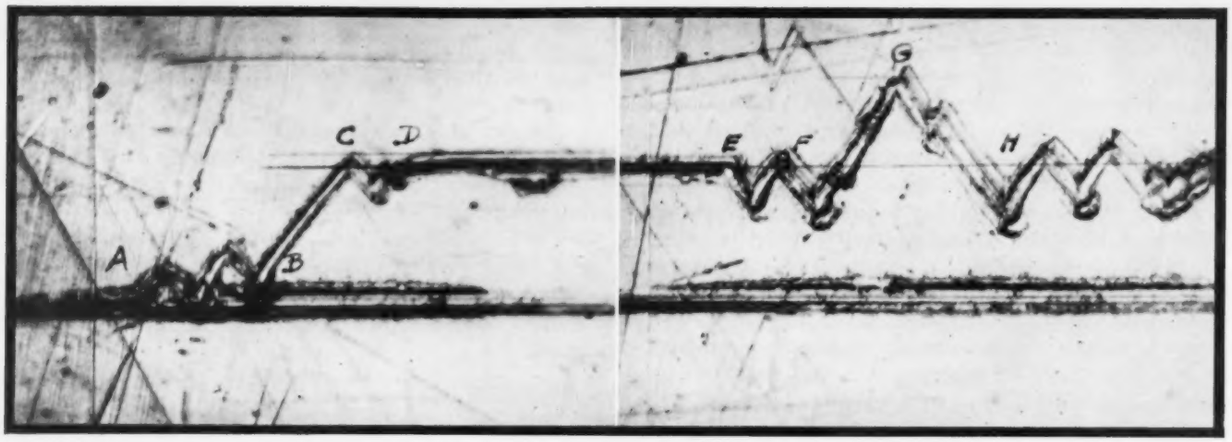


Figure 1

of tests was made on the crane located in the Fritz Engineering Laboratory. This, a 10 ton Niles crane No. 10097, has a span of 47 ft., and is a riveted fish-belly box girder. The bridge brakes are foot-operated mechanical brakes. The braking tests were made with no load on the crane due to limited clearance in the laboratory. A number of impact tests were also made on this crane. The jerk impact tests were made in the following manner. Slack was let out of the hoist. Then the slack was taken up so that the hook was moving at the maximum hoisting speed when the load was jerked off the floor. A tank of water weighing 8250 lb. was used for the load. To simulate bad joint impact, the crane was run over ¾ in. oak wedges placed under both ends of one bridge girder. Under the weight of the crane, the wedges flattened down to 9/16 in. and this was assumed to be the drop of the crane. The 8250 lb. load was also used in the wedge tests and the crane was tested with 21 ft. and 14 ft. distances between the hook and hoisting drum.

Three cranes were tested at the Bethlehem plant of the Bethlehem Steel Corporation. The first crane, Bethlehem No. 430, was a new 10 ton riveted, fish-belly, box girder of 69 ft. span with hydraulic bridge brakes. Since the shop was under construction, the full length of the runway was available for test.

In the lateral tests, the load was picked up with the crane stationary. The crane was then fully accelerated until maximum speed was attained, and then the brakes were fully applied. These tests were made with a 31,000 lb. load of billets hung first as high as possible and then just high enough to clear the floor. Similar tests were made with a load of 15,500 lb. The clearance from the floor to the trolley rail of this crane was 23 ft. 9 in.

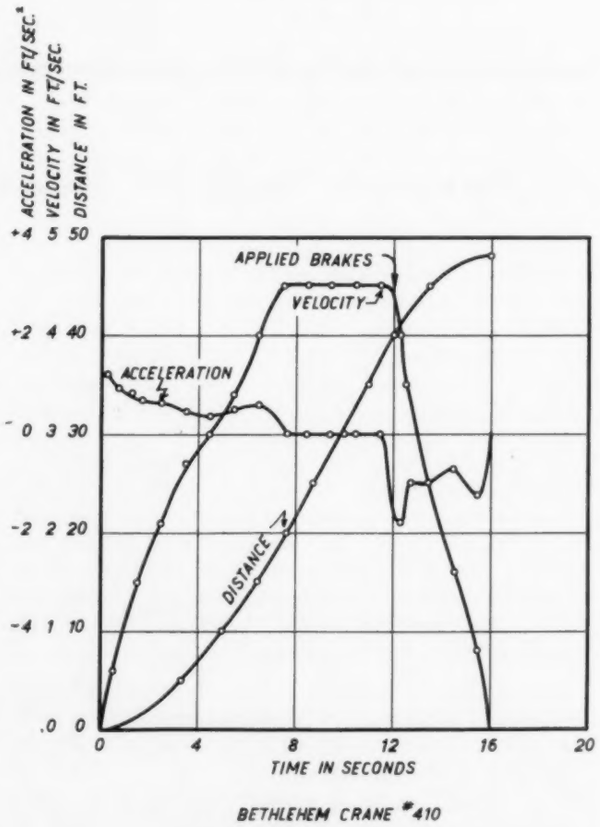
Jerk impact tests were also made on this crane. The load was raised with a jerk, and it was stopped with a jerk while being lowered.

The second crane tested at the Bethlehem plant was Bethlehem No. 410. This crane, located in the electrical shop, is a 5 ton hand-operated I-beam crane of 37 ft. 4 in. span. Lateral load tests were made on this crane using a magnet weighing 9100 lb. for the vertical load.

The third crane tested was Bethlehem No. 4. This is a 30 ton riveted, fish-belly box girder crane of 75 ft. span which is used as a skullcracker. This crane was tested to determine what impact is present as the ball is released from the magnet. A ball weighing 19,500 lb. was used in the tests and the magnet weighed 8800 lb.

The stresses in the cranes were measured by means of the De Forest scratch recording strain gauge. Since the stresses in the cranes were too small to use the gauge as manufactured, a special holder was made to increase

Figure 2



SPEED-VELOCITY CURVES

TABLE I

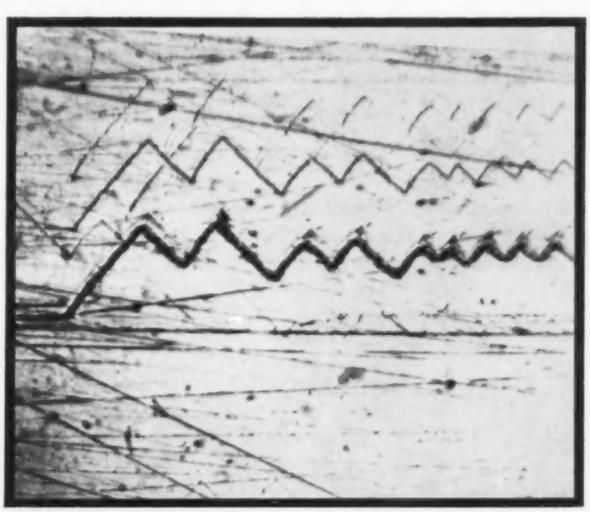
Lateral Load Tests

Crane	Test No.	Vertical load, lb.	Static stress, lb. per sq. in.		Maximum acceleration, ft. per sec.²	Measured deceleration, ft. per sec.²	Maximum speed, ft. per min.	Measured lateral stress, lb. per sq. in.		Computed lateral stress using 10 per cent lateral load, lb. sq. in.	Lateral load from measured braking, per cent
			Computed	Measured				Accel.	Braking		
Lab.	1	Dead load only	0	0	2.0	4.0	360	...	800	660	12.5
Beth. 410	1	9,100	2000	2050	1.2	1.8	180	800	1000	2500	5.6
Beth. 430	1	31,000	2500	1800	2.6	3.2	390	800	950	1300	10.0
Beth. 430	2	31,000	2500	1700	2.6	3.2	390	750	700	1300	10.0
Beth. 430	3	15,500	1250	1100	2.6	3.2	390	650	700	960	10.0

the gauge length to 20 in. Pictures of the strains were taken with a microscope using a magnification of 450X. Figure 1 shows such a picture. The gauges were all calibrated against a 20 in. Whittemore strain gauge. This was done in the laboratory by measuring the strains on a beam under a static bending load. In the crane tests, the gauges were mounted on both edges of the top and bottom flanges of the crane girder at the span center. The trolley was also located at the span center during the tests. Occasional difficulty was encountered with some of the gauges sticking so that no scratch was recorded, but generally enough gauges worked so that the lateral stresses could be determined. In addition each test was repeated to insure the availability of enough records to determine the dynamic stresses.

The acceleration and deceleration were determined from stop-watch readings measuring the time to one-tenth of a second. Distances were marked on the runway rail and the time at each mark was found from the stop-watch. From these measurements, the speed and acceleration can be easily determined. Figure 2 gives a picture of the acceleration and deceleration of Bethlehem, No. 410. This method is not as accurate as it should be, and in future tests it is suggested that a moving picture camera be used to photograph the stop-watch and distance marks along the rail at the same time. The acceleration and deceleration, as measured, apply only to the bridge of the crane and do not necessarily apply to the load since the motion of the load is modified by the swing.

Figure 3



TEST RESULTS

About one hundred pictorial records of dynamic stresses were taken in this investigation. Typical records are illustrated in this report and a summary of the pertinent data is given in Tables I and II.

Table I summarizes the results of the lateral load tests on the several cranes. The columns of computed and measured static stress apply to the vertical live load. The static stresses in the tables are the arithmetical average of the tension and compression flanges. The acceleration, deceleration, and speed in the table are taken from curves similar to Figure 2. The measured lateral stresses are the arithmetical average of the readings on all the flanges, and are due to the live load and the dead load. The column of computed lateral stress is found by assuming that 10 per cent of the live load, trolley, and dead load acts laterally. The last column (lateral load from measured braking) is obtained by dividing the observed deceleration by g, the acceleration

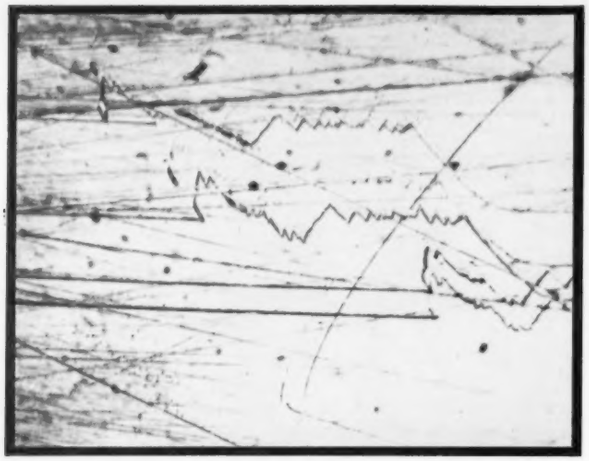


Figure 4

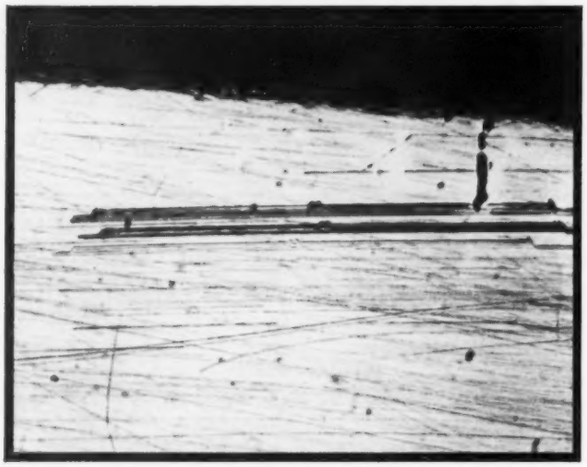
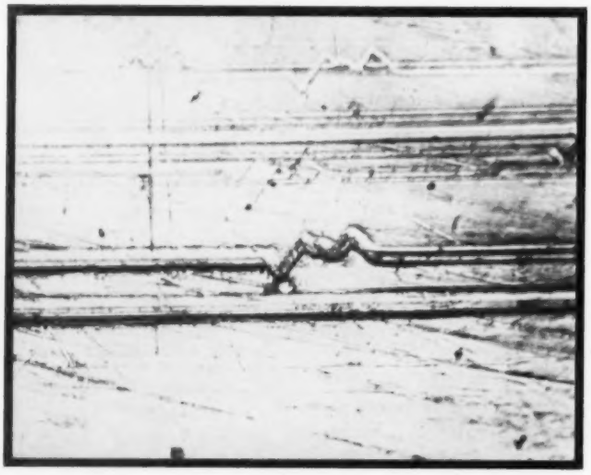


Figure 5

▲

Figure 6



of gravity, and multiplying by 100 to get the percentage.

With the exception of the I-beam crane, the observed deceleration is about 10 per cent of gravity as shown in the last column of Table I. However, the measured stresses are somewhat less than those which correspond to the observed braking force except for the laboratory crane.

The acceleration and deceleration stresses for the I-beam crane were somewhat larger on the top flange than on the bottom flange. The average value is given in the table.

Table II gives the results of the impact tests on the various cranes. The static stress referred to in this table is that due to the live load. For the jerk impact test, this is the only force which causes an impact stress. However, when the crane is run over a bad joint or wedges as in the case of the laboratory crane, there is an impact due to the weight of the trolley and girders, in addition to the live load impact. For this reason, two columns are given for the per cent of impact, the first column is the per cent impact based on the live load, and the other is the per cent impact based on all the loads causing impact. In specifications most impact factors are based only on the live load.

The variation in the impact factor due to changing the length of the hoist is shown in tests 3 and 4 on the laboratory crane, where the impact decreased from 100 to 56 per cent of the live load with an increase in the length of the hoist.

The jerk impacts varied from 9 to 33 per cent.

Bethlehem No. 4 is a skullcracker crane, on which the impact was measured as the ball was released. The measured impact stress was 37 per cent of the weight of the ball and 25 per cent of the weight of the ball and magnet, as found from the scratch records. The center deflection was also measured in this crane with a 0.001 in. Ames dial and the impact measured from the dial was 20 per cent of the ball and magnet. This value may be a little low due to inertia of the dial. The jerk impact (not given in the table) was 12 per cent of the ball and magnet as found from the deflection.

Figure 1 shows the scratch record for one edge of the compression flange in one of the wedge impact tests. All distances in this picture are measured from the centerline of the scratch. The portion AB shows the whipping action of the hoist as the cable tightens. In BC the load is being transferred to the crane. Between C and D the jerk impact can be seen. The distance between the bottom line which is the line of zero stress and the top line DE represents the static stress. At E the crane dropped off the wedges, and the reduction in stress, while the crane is dropping is shown in this portion. Before this stress has been completely released, however, the crane hits the runway rail and point G gives the sum of the static stress plus maximum impact stress. The portion of the curve after H shows the vibrations in the crane after impact.

Some other scratch curves are given for illustration in Figures 3, 4, 5, and 6.

In Figure 3 is shown the static stress and vibration which occur when the load is picked up quickly. The variation of the vibrations about their neutral position is a measure of the jerk impact.

In Figure 4, are shown the acceleration and decelera-

TABLE II

Impact Tests

Crane	Test No.	Type of test	Load	Static stress, lb. per sq. in.		Measured impact stress, lb. per sq. in.	Impact in per cent of:		Distance from hoist drum to hook, ft.	Measured impact from deflection
				Computed	Measured		Computed live load stress	Total load acting in impact		
Lab.	2	Jerk impact	8,200	1530	....	460	33.0	33.0	21.0	...
Lab.	3	*	8,200	1530	1610	850	56.0	31.0	21.0	...
Lab.	4	*	8,200	1530	1400	1400	100.0	51.2	6.0	...
Lab.	5	Jerk impact	8,000	1500	1340	140	9.3	9.3	21.0	...
Lab.	6	Jerk impact	8,000	1500	1500	180	12.0	12.0	21.0	...
Lab.	7	Jerk impact	8,000	1500	1500	180	12.0	12.0	21.0	...
Beth. 4	1	Impact	19,500	1300	1500	475	36.6	25.0	4.0	20.0
Beth. 430	1	Jerk impact	31,000	2500	1800	550	22.0	22.0	16.0	...

*Crane run over $\frac{5}{16}$ in. wedges.

tion stresses for Bethlehem No. 430 in one of the lateral load tests.

Figure 5 is a low-power magnification of the scratch record of Bethlehem No. 430. The left portion shows the stress as the load is lifted. The center shows the acceleration and braking stresses, while the right end shows the release of stress when the load is taken off the crane. Figure 6 shows the lateral stresses of Figure 5 under a higher magnification.

DISCUSSION OF RESULTS

The column for maximum force of deceleration of Table I shows that a value of 10 per cent of the vertical load gives a fair approximation of the lateral load. However, the measured stresses for the Bethlehem cranes are somewhat less than the computed strains. One reason for this is that the computations neglected the walkway and end fixity. Since there were a walkway and end ties on these cranes, these factors probably reduced the lateral stresses. This would be particularly true for Bethlehem No. 430 which was a new crane with a tight end connection. The ratio between the observed lateral stresses and those computed for the bridge girder as a simple beam, based on the observed maximum braking force, vary from 53 to 73 per cent.

The laboratory crane which is old and has practically no end connection would be expected to show no end fixity and in this crane the measured and computed stresses were in fair agreement.

A value of 10 per cent for the lateral load would correspond to a coefficient of friction of 0.20 if one-half the bridge wheels were braked. If all the wheels were braked, the lateral force would be 20 per cent. The value of 0.20 for the coefficient of friction is one which would not be at all unlikely under actual operating conditions.

The lateral end fixity probably can not be taken into account in design, since the end connections on a crane would loosen in service, and reduce the fixity.

On the other hand, some fixity will probably be present at all times, no matter how much the end connection loosens. Any end fixity present however, will provide an additional safety factor if the crane is designed as a simple beam for the lateral load.

The fact that the maximum swing of the load and the maximum acceleration may not have occurred at the same time may also have reduced the observed stress. This can be seen in Figure 2 where it will be noted that there are two peaks in the deceleration curve. These peaks can also be seen in the acceleration and braking stresses in Figure 4. This effect is due to the fact that when the tangent of the angle of swing is greater than the coefficient of friction, the load is being decelerated more than the bridge, and when this angle is less than the coefficient of friction the load is being decelerated less than the bridge. In actual practice, however, a good crane operator will operate his brake in such a manner as to keep the swing down to a minimum.

The impact stresses caused by running the crane off wedges under both ends may represent an unduly harsh condition, if this test is to simulate bad runway joints; because bad joints under both ends of the girder are unlikely to occur at the same time. Further tests, with wedges under only one end of the bridge should be made. In the tests made, this type of impact was the most severe, and it is quite likely that it is the type of impact which should govern in design. The wedge impact tests showed also that this impact is much more severe when the load is close to the hoist drum. Since this is the position in which the load is usually placed as the crane moves down the runway, it should govern in design. Impact due to bad joints is also particularly severe, since the weight of the trolley and girders also contribute to the impact stresses, which is not the case in other types of impact. Whenever the length of hoist is great, the impact will be reduced because of the cushioning effect of the cables.

The jerk impact was a smaller factor than that due to dropping the crane off wedges. If this, then, is always the smaller factor, the design impact factor should not be a function of the hoist speed of the crane.

One interesting effect was noted in the impact test on

the skullcracker. After the switch was cut to release the ball, it took an average of 1.2 seconds before the ball left the magnet. This was apparently due to residual magnetism in the magnet even though the switch was of such a type that a reverse current was applied.

CONCLUSIONS

The dynamic tests discussed in this report indicate that for the cranes tested:

1. The maximum lateral force on a crane was due to braking and was approximately 10 per cent of the vertical forces.
2. Some end fixity was present which reduced the stresses due to the lateral force. However, this fixity probably can not be counted on in design and the full value of 10 per cent for the lateral load should be used in design.
3. Jerk impact, as measured, varied from 9 to 33 per cent.
4. The impact due to running the crane off wedges to simulate bad joints was quite large. For the tests made, values of 56 and 100 per cent of the live load were measured. More tests on this factor are needed.
5. The closer the load is to the hoist drum, the greater will be the impact.

4. Torsional Properties of Fabricated I Beam and Box Sections

▲ THIS REPORT gives the results of a series of torsion tests on riveted and welded I-beams, and riveted and welded box girders. Due to the slipping which occurs in the seams, the fabricated girders were neither as strong nor as rigid as the equivalent girder section without longitudinal seams.

On the basis of the tests which were made, it was found that a riveted section was about one-third as stiff in torsion as a section in which no slipping occurs. Design methods for computing the stresses and twists of built-up girders are included in this report.

The stresses and twist in a hollow tube may be predicted fairly accurately by the theoretical formulæ proposed by R. Bredt. These formulæ state that the average shear stress in the walls of the tube due to torque is:

$$\tau = \frac{M}{2At}$$

The angle of twist per unit length is:

$$\Theta = \frac{M}{GJ}$$

τ is the shearing stress in pounds per square inch, M is the twisting moment in inch-pounds, t is the wall thickness of the tube, A is the area bounded by the center lines of the tube walls, G is the shear modulus (about 11,500,000 lb. per in.^{2} for steel), and J is the torsional constant in in.^{4}.

J is analogous to the polar moment of inertia and for a hollow tube:

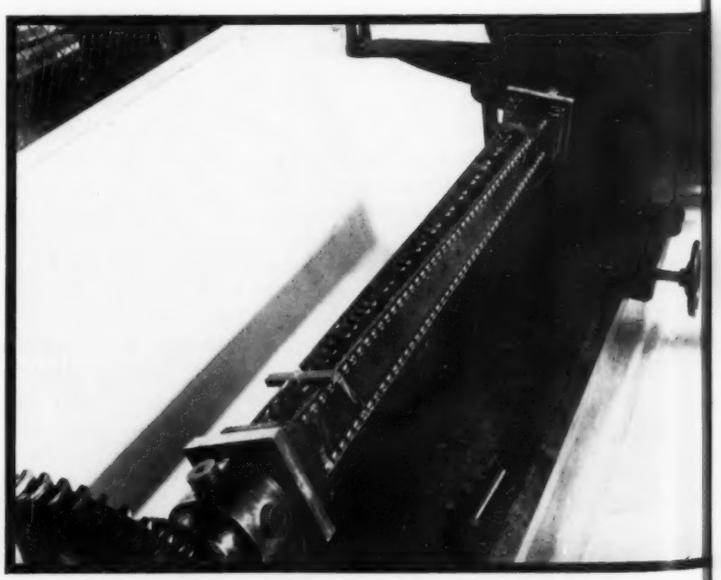
$$J = \frac{4A}{\Sigma \frac{s}{t}}$$

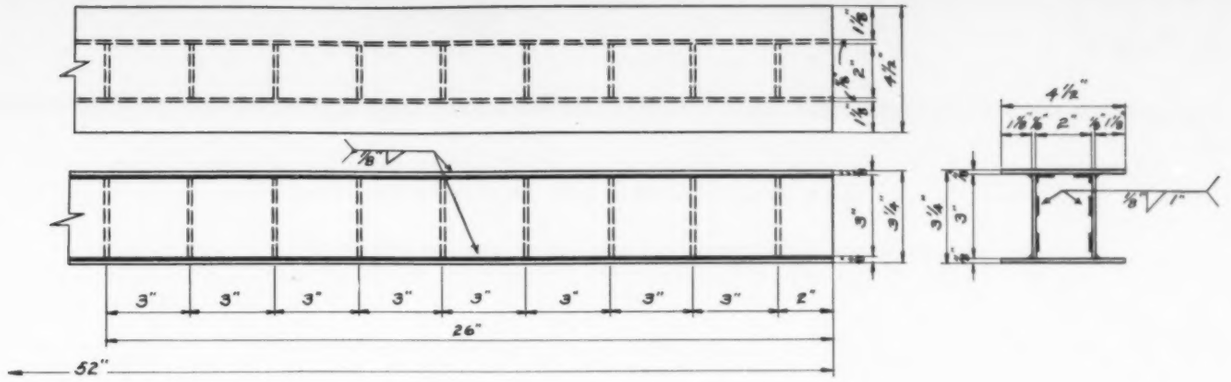
where s and t are the length and thickness of each side.

The above formulæ apply to hollow tubes of all shapes which are bounded by one single line. The formulæ are approximate, but give close results when the thickness of the tube wall is small in comparison with the distance of the wall from the center of gravity of the tube. For crane girders this approximation is usually small.

The formulæ are derived for one piece tubes, thus they do not necessarily apply to built-up sections since slipping may occur along the longitudinal seams. The torsional strength of a built-up section, loosely bolted together, would be only the sum of the strengths of the individual pieces. This sum is but a fraction of the

Figure 2





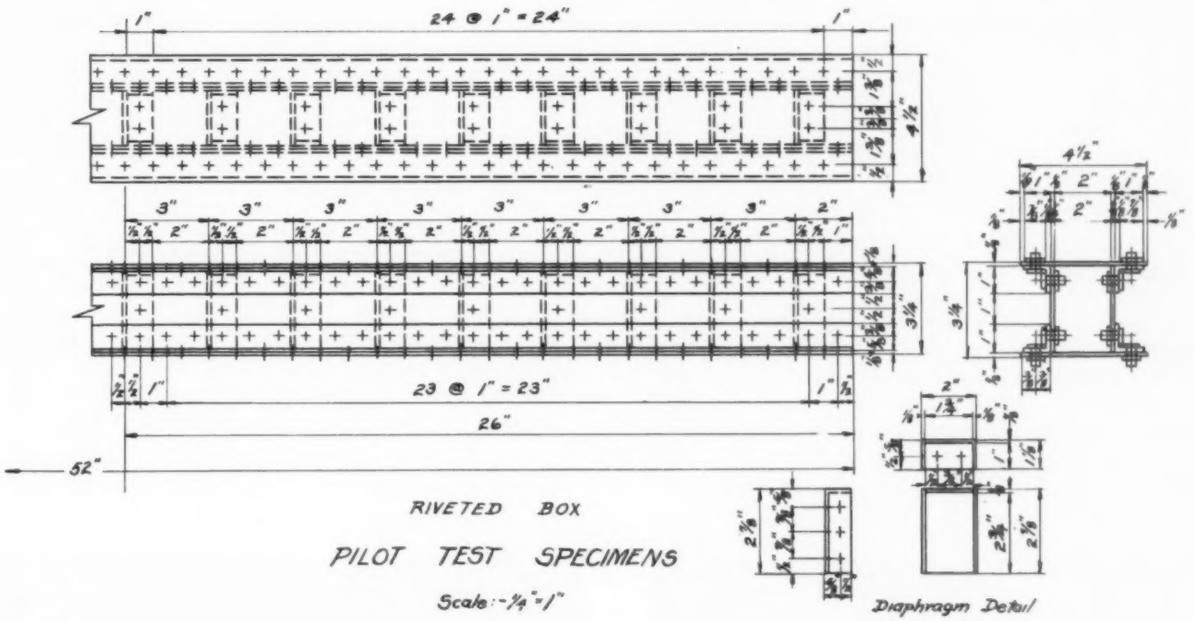
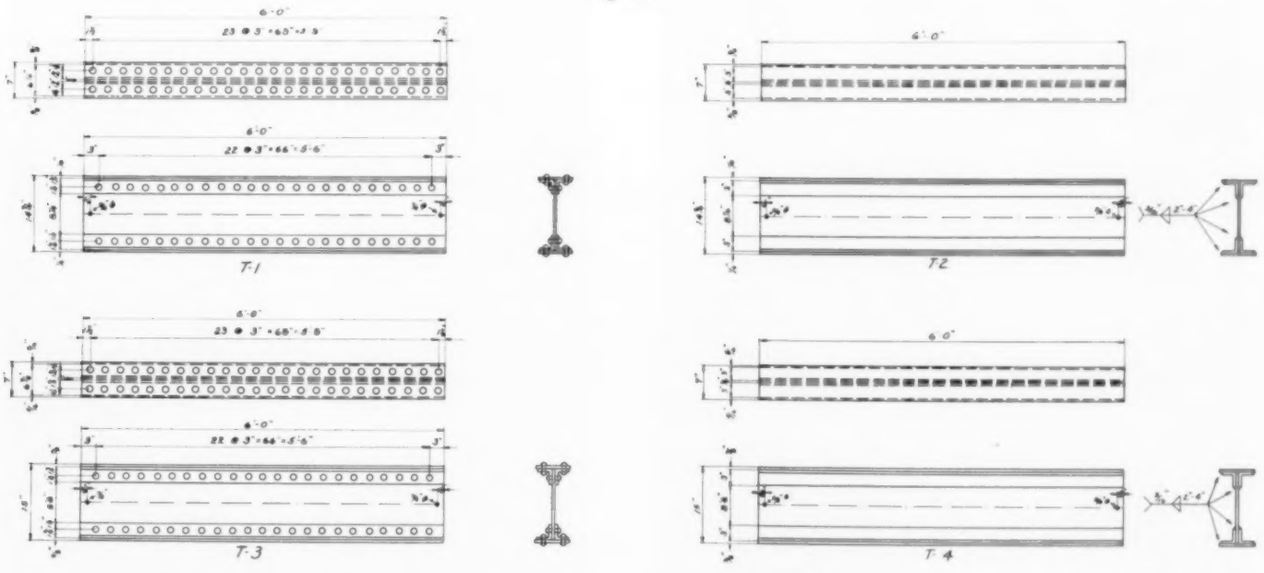


Figure 1

▲

Figure 3



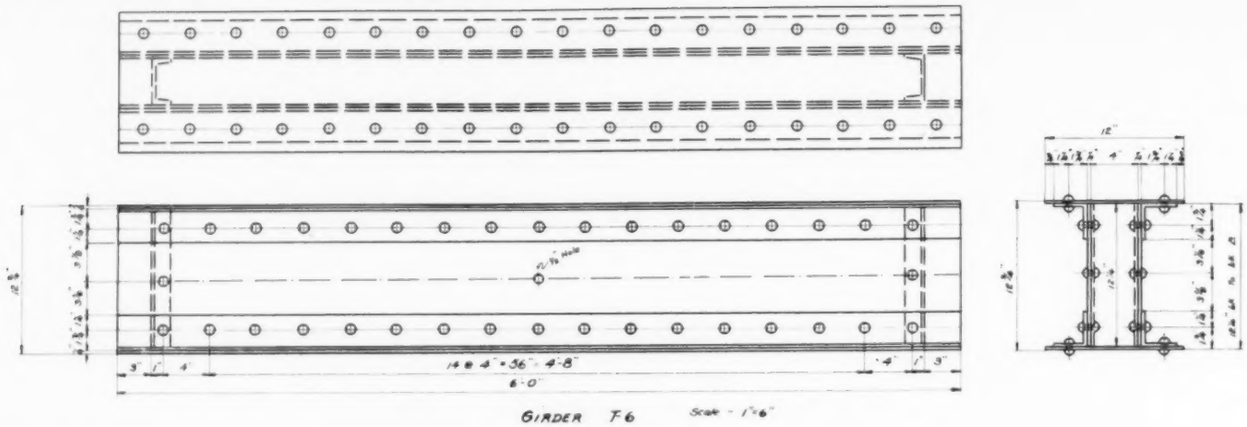


Figure 4

strength of the section as a tube. In a continuously welded section, on the other hand, no slipping can occur along the seams and the twisting strength should equal that of the solid tube as predicted by the theory. A built-up riveted or tightly bolted section should have properties somewhat less favorable than the welded or solid tube since a limited amount of slip may occur.

This series of tests was made to determine how the actual behavior of built-up box sections compared with that of one-piece sections. In addition, built-up I-beams were also tested to compare their behavior with solid I-beams, for which rational methods of analysis are available. ("Structural Beams in Torsion," Inge Lyse and Bruce G. Johnston, *Transactions A.S.C.E.*, 1936, p. 1389; *Bethlehem Manual of Steel Construction*, p. 279.)

Pilot tests were made on a series of small 2 by 3 in.

Figure 5

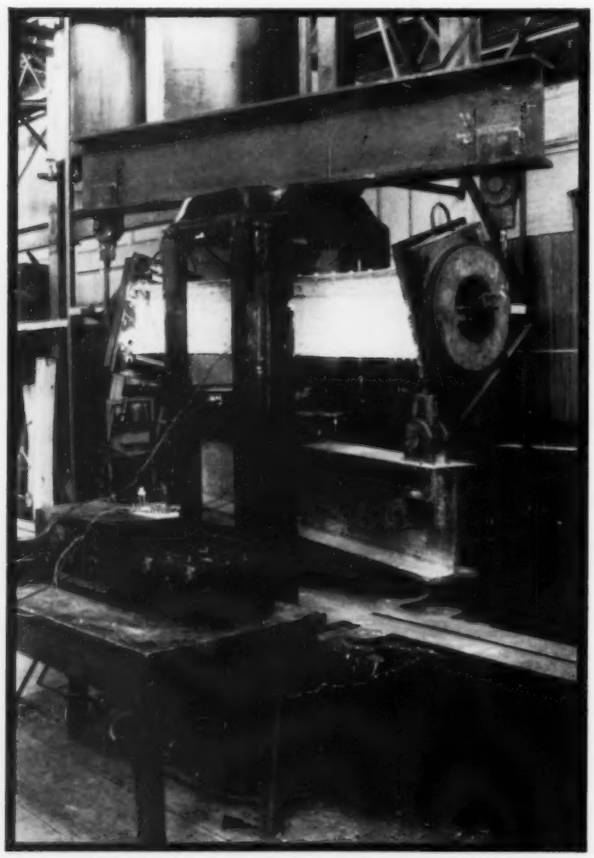
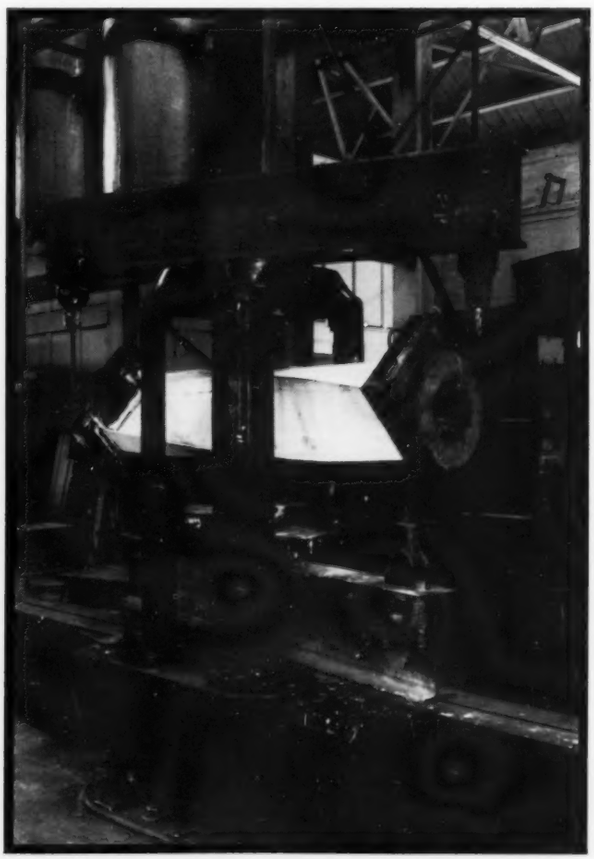
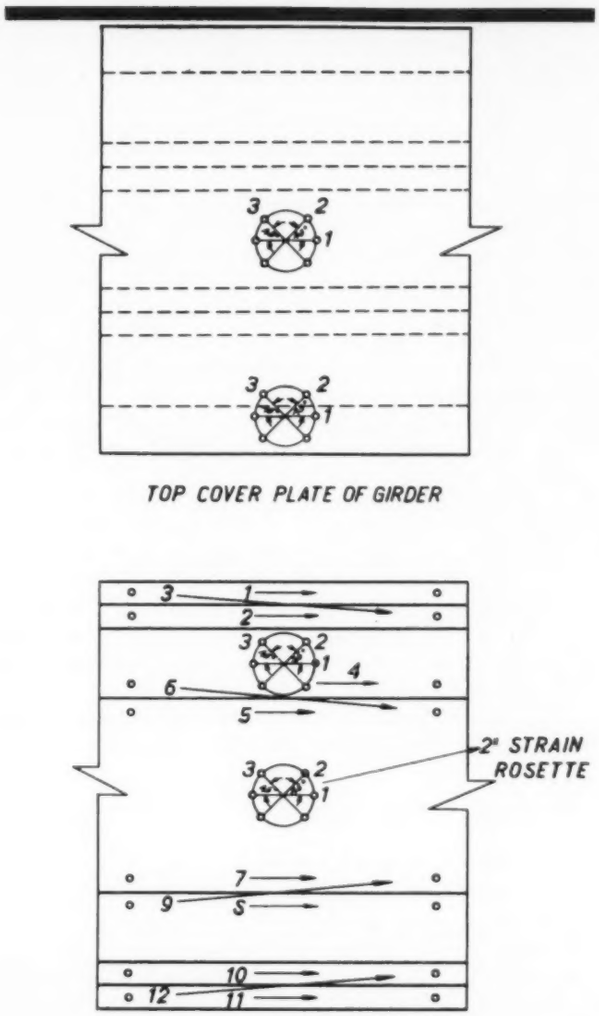


Figure 6



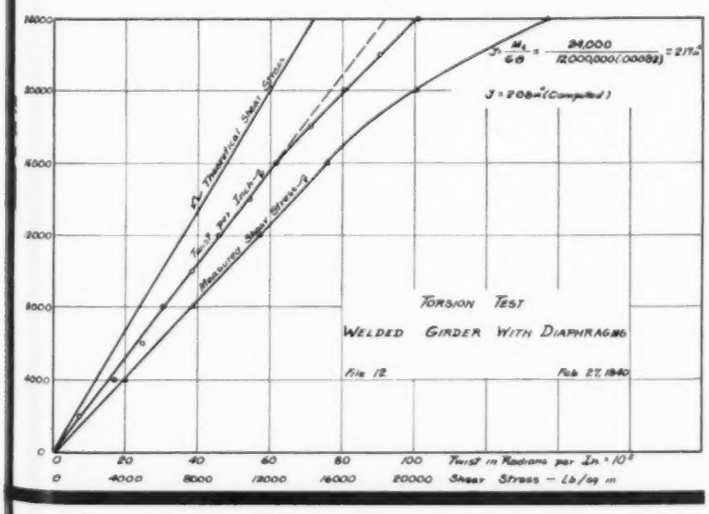


LOCATION OF STRAIN GAGE LINES

Figure 7

▲

Figure 8



box sections, 50 in. long and made of ⅛ in. plate. The sections consisted of welded and bolted boxes as shown in the drawing of Figure 1. These boxes were tested in a torsion machine of 24,000 in.-lb. capacity. Figure 2 shows a picture of the bolted box being twisted in the testing machine.

The bolted girders were made to simulate the action of a riveted girder by using ⅜ in. bolts which fitted tightly in the holes. Tests were made on the bolted girder with the diaphragms fastened on one, two, and three sides, and with diaphragm spacings varying from 3 to 48 in.

Pilot torsion tests were also made on two welded boxes, one had diaphragms as shown in Figure 1, the other had no diaphragms. Three tests were made on the box with the diaphragms. It was first tested with the section shown in Figure 1, then it was tested with flange angles welded to the box to make the section similar to a riveted section. Finally, the top plate and angle were burned off, leaving a plain rectangular box, and the torsional constant was again found.

The welded box in the pilot test behaved as the theory predicted, so further torsion tests were not deemed necessary on the welded sections. However, the twist and the stresses in the bolted box were so much higher than that predicted by theory, it was thought advisable to make additional tests on riveted sections. Tests were made, therefore, on four I-beam and three box sections. The details of the I-beams are shown in Figure 3. The details of the box girder called T-6 is shown in Figure 4. The other box girders were similar to T-6 except for the diaphragms. T-5 had no diaphragms and T-7 had four diaphragms equally spaced. The riveted girders were fabricated with ½ in. rivets, which were designed to take the longitudinal shear of the girders under a concentrated center load.

Since these specimens were too large to be tested in an ordinary torsion machine, they were loaded by a special rig attached to a 300,000 lb. Olsen machine. ("Torsion Testing Machine of 750,000 Inch-pound Capacity," Bruce G. Johnston, *Engineering News-Record*, Vol. 114, No. 9, p. 310, February 28, 1935.) Figures 5 and 6 show the details of the load rig and also show girders T-2 and T-6 being tested. The girders were free to warp at the ends except as the warping might be influenced by frictional forces at the ends.

The twist in the girders was measured by means of level bars. The strains, from which the stresses were determined, were measured with a 2 in. Olsen and a 10 in. Whittemore strain gauge on the outside of the boxes and both sides of the I-beams. The strains were measured longitudinally along the girder and on strain rosettes located on the webs and flange. The shear stresses were found from the strain rosettes. So far as the author knows, very few tests have been made in which the stresses have been measured in box beams under torsional loads. ("Torsion of Rectangular Tubes," William Hovgaard, *Journal of Applied Mechanics*, September 1937, p. A-131.)

The slip in the seams of the girder was measured by the Whittemore gauge with one end of the gauge on one side of the seam and the other end on the other side of the seam. The locations of the gauge lines are

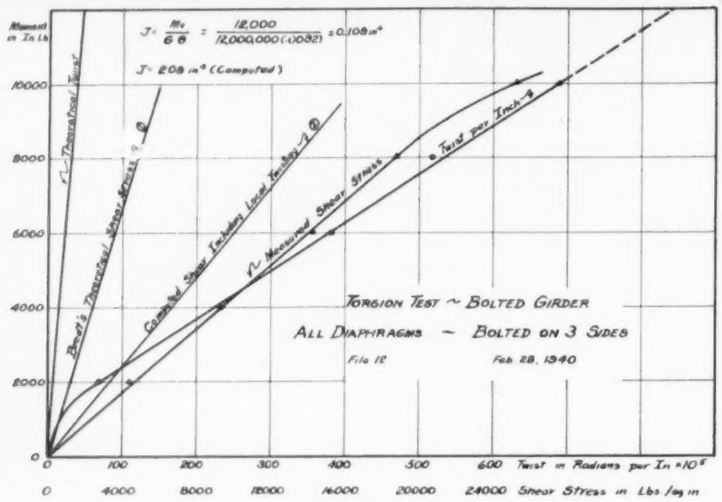

Moment in In Lb
J = Mx/θ = 12,000/12,000,000(.0082) = 0.108 in⁴
J = 2.08 in⁴ (Computed)
Bredt's Theoretical Twist
Bredt's Theoretical Shear Stress
Combined Shear Tendency Local Tendency = 1.0
Measured Shear Stress
Twist per Inch = θ
TORSION TEST ~ BOLTED GIRDER
ALL DIAPHRAGMS ~ BOLTED ON 3 SIDES
File 12
Feb 28, 1940
Twist in Radians per In ×10⁴
Shear Stress in Lbs/sq in

Figure 9

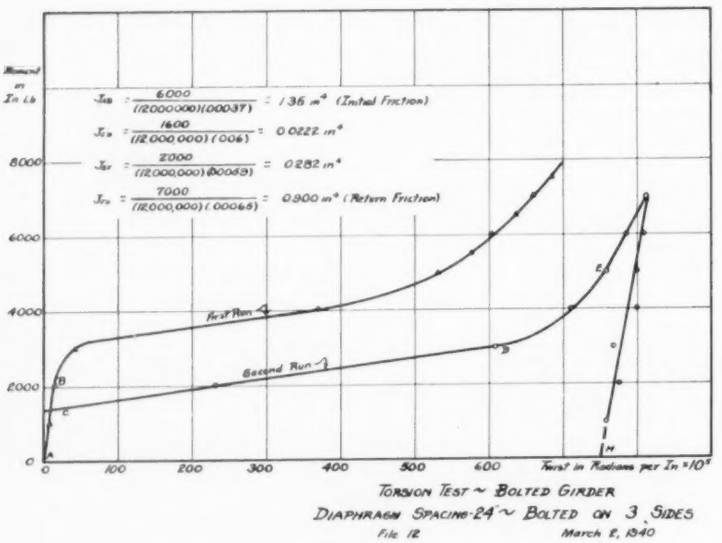
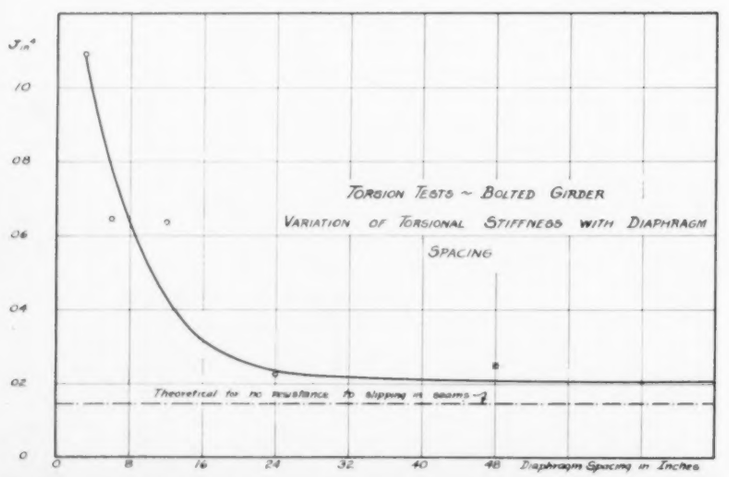

Moment in In Lb
J = 6000/(12000000)(.00037) = 1.35 in⁴ (Initial Friction)
J = 1600/(12,000,000)(.006) = 0.0222 in⁴
J = 2000/(12,000,000)(.00589) = 0.282 in⁴
J = 7000/(12,000,000)(.00645) = 0.900 in⁴ (Return Friction)
First Run
Second Run
Twist in Radians per In ×10⁴
TORSION TEST ~ BOLTED GIRDER
DIAPHRAGM SPACING 24 ~ BOLTED ON 3 SIDES
File 12
March 8, 1940

Figure 10

▲

Figure 11


J in⁴
TORSION TESTS ~ BOLTED GIRDER
VARIATION OF TORSIONAL STIFFNESS WITH DIAPHRAGM SPACING
Theoretical for no resistance to slipping in seams
Diaphragm Spacing in Inches

shown in Figure 7. The gauge lines marked 3, 6, 9, and 12 are a measure of the slip when corrected for the actual strain in the elements themselves.

TEST RESULTS

Pilot Tests—Table I presents the results of the twisting tests on the pilot specimens. The torsion constants of the welded boxes agree with the theoretical values of 2.09 in.⁴. The flange angles and outstanding portion of the coverplate have little effect on the results. Neither did the diaphragms have any appreciable effect, since in these girders no slipping occurred to bring the diaphragms into action.

The values of the torsion constants, as shown in Table I, are quite variable for the bolted girder. They are much less than that given by Bredt's theory for a box. This is due to the slipping which occurred along the seams of these girders. As the table shows, the diaphragms had an important effect on the twisting of the beam. When the diaphragms were fastened on three sides, they were much more effective than when fastened on two sides. Secondly, increasing the number of diaphragms, when bolted on three sides, increased the stiffness of the girders.

Figure 8 gives the results of the torsion test on the welded box. The outside stresses, as measured, are

TABLE I

Welded Girder

Measured values of J in.⁴	Remarks on test
2.17	3 in. diaphragm spacing
2.25	3 in. diaphragm spacing (Flange angles added)
2.03	3 in. diaphragm spacing (Flange angles cut off)
2.09	No diaphragms
2.09	Theoretical value by Bredt's theory

Bolted Girder

Measured values of J in.⁴			Remarks on test
Slipping	Built-up	Friction	
.0391	.242	...	3 in. diaphragm spacing Diaphragms bolted on 1 side
.0393	.445	...	3 in. diaphragm spacing Diaphragms bolted on 2 sides
.1090	...	...	3 in. diaphragm spacing Diaphragms bolted on 3 sides
.0647	.180	...	6 in. diaphragm spacing Diaphragms bolted on 3 sides
.0639	.229	...	12 in. diaphragm spacing Diaphragms bolted on 3 sides
.0222	.282	1.35	24 in. diaphragm spacing Diaphragms bolted on 3 sides
.0250	.131	1.32	48 in. diaphragm spacing Diaphragms bolted on 3 sides
.0204	.264	1.02	No diaphragms
.0146	...	...	Theoretical value for freely slipping bolts

TABLE II

Girder	Twist, Θ in radians per in.	Shear in web		Shear in flange		Direct stress in web, 4 in. from center line		Direct stress, edge of flange		
		Measured	Computed $\tau = tG\Theta$	Measured	Computed $\tau = tG\Theta$	Measured	Computed	Angle, measured	Cover plate, measured	Computed*
T-1	.0057	16,800	17,100	26,600	34,200	−12,600	−10,300	+11,500	+16,400	+12,500
T-2	.0040	12,800	12,000	16,000	24,000	− 3,600	− 4,900	+ 9,500	+ 6,000	+ 6,000
T-3	.0024	13,400	10,800	16,000	21,600	− 1,800	− 1,800	+ 3,200	+ 3,700	+ 2,100
T-4	.0014	6,200	6,300	7,600	12,600	− 1,200	− 600	+ 800	+ 700	+ 600

All values given for torque load of 40,000 in.-lb.

*Formula for direct stress at edge of flange:

$$f = E\Theta^2\left[\frac{h^2}{8}+\frac{w^2}{8}-\frac{th^3+6h^2tt'+2t'w^3+2t''h^3-16t''a^3}{24(ht+2t'w+ht''-4at'')}\right]$$

Formula for direct stress in web:

$$f = E\Theta^2\left[\frac{y^2}{2}-\frac{th^3+6h^2tt'+2t'w^3}{24(ht+2t'w)}\right]$$

h = height of web
t = web thickness
t' = flange plate thickness
w = width of flange
t'' = thickness of flange angle
a = distance to bottom of flange angle
y = distance to point in web where stress is computed
Θ = angle of twist in radians

about 60 per cent grester than the theoretical average as computed by Bredt's theory. The twist checks closely. Figure 9 gives a similar curve for a bolted girder. Two computed shear stresses are shown in this figure. The lower stress curve marked 1 is that given by Bredt's theory, while the higher value marked 2, includes the shear stress due to the twisting of each side of the box. This second factor is small for a box with no slip and can be safely neglected. It is large, however, in the present case.

Figure 10 gives the results of the torsion test on the bolted girder with the diaphragms spaced at 24 in. It shows in detail the action of the girder. When the girder is first twisted, as shown in the curve marked "First Run," the box is initially very stiff due to the friction of the seams. However, as the load is increased, the joints slip, and the friction is gradually broken. The portion of the curve marked *CD* shows the twist which takes place while the bolts are slipping. Most of the total twist occurs in this range. At point *D*, the bolts start to bear and the portion of the curve beyond *D* shows the increased resistance to slip which results. At point *F*, the load was released, and the section *FH* on the curve shows the effect of the return friction in preventing the girder from untwisting. If the testing machine had been of such a type that the box could have been twisted in the reverse direction, it is expected that the resultant curve would have a large hysteresis loop.

Figure 11 shows the variation in the torsional stiffness as the number of diaphragms is increased.

I-Beam Tests—The test results on the I-beams were quite similar. T-1 and T-2 were identical in section but T-2 was welded and T-1 was riveted. T-3 and T-4 were similar to T-1 and T-2 except that the sections of the former were $\frac{3}{8}$ in. thick, and the sections of the latter were $\frac{1}{4}$ in. thick.

Since T-2 and T-4 were welded, less slip could occur. Therefore, the stresses and twists in these girders should be less than those of girders T-1 and T-3.

Figure 12 compares the twists of the four I-beam girders under a torque load. The rotation given is the average of the rotation of the flanges and the webs except for T-3 for which the top twist is given. The top twist of T-3 is somewhat less than the average, since the web usually twisted a little more than the flange. The values for the torsion constants are appreciably less than the theoretical values for the equivalent solid I-beam.

Figure 13 gives the magnitudes of the shear stresses in the web and coverplate of the four I-beam girders. This figure shows that in the same girder section, as for example T-3 and T-4, the measured stresses in the riveted section are about twice that of the welded section. This is the result of the greater twist of the riveted girder. The shear stresses in the flange are greater than those in the web. Theoretically, disregarding stress concentrations, the flange shearing stresses should be twice the web stresses for the girders tested since the flange is twice as thick as the web.

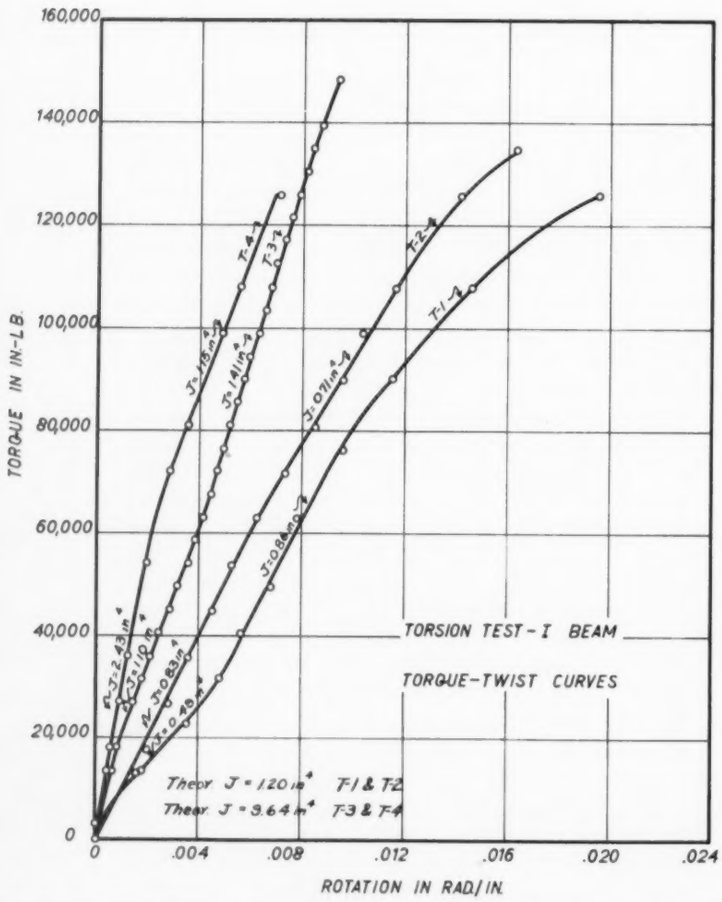
Table II gives a comparison of computed and measured stresses for the 40,000 in.-lb. load. The shear stresses were computed by the formula $\tau = tG\Theta$, where τ is the shearing stress, and t is the thickness of the section considered. This formula comes directly from the formulæ for twist and stress of a narrow rectangular section. The web stresses check closely as shown by the

table. The flange stresses are larger than the web stresses but smaller than the theoretical. This shows that some slippage is taking place and that the full thickness of the flange can not be assumed to be acting as a unit.

Figure 14 gives curves for the variation of the slip with the twist and the moment for girders T-1 and T-2. Similar results were obtained for T-3 and T-4. The slip given is the total of the slips in all the seams around the girder. It should be noted that for T-1, the twist is proportional to the slip. The slips for T-2 are much smaller than for T-1. The values for T-2 are not actually slips since the seam was welded, but since the welding was intermittent, the welds are strained higher than the adjacent plates and this results in a relative strain between the plates.

Secondary longitudinal direct stresses are set up in a twisted I-section or box as a result of the tendency of the fibers farthest from the center of twist to be lengthened. In Figure 15 are plotted the curves of direct stress in T-1 as actually measured. Table II gives a comparison between computed values of the stress and the measured values at the 40,000 in.-lb. load. These stresses are quite appreciable. The formula for the direct stress in the I-beam was derived in a manner

Figure 12



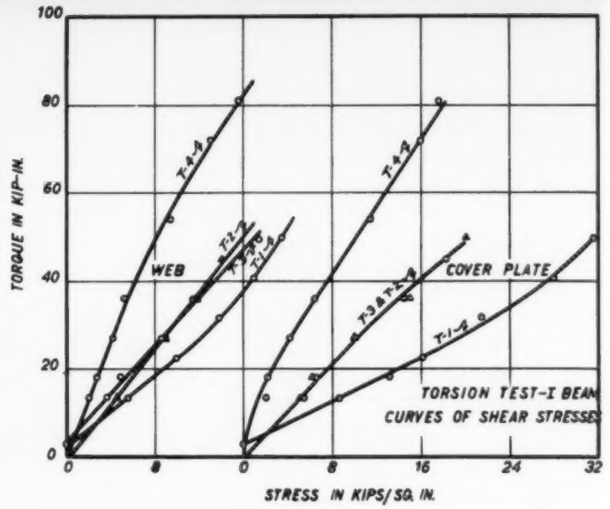


Figure 13

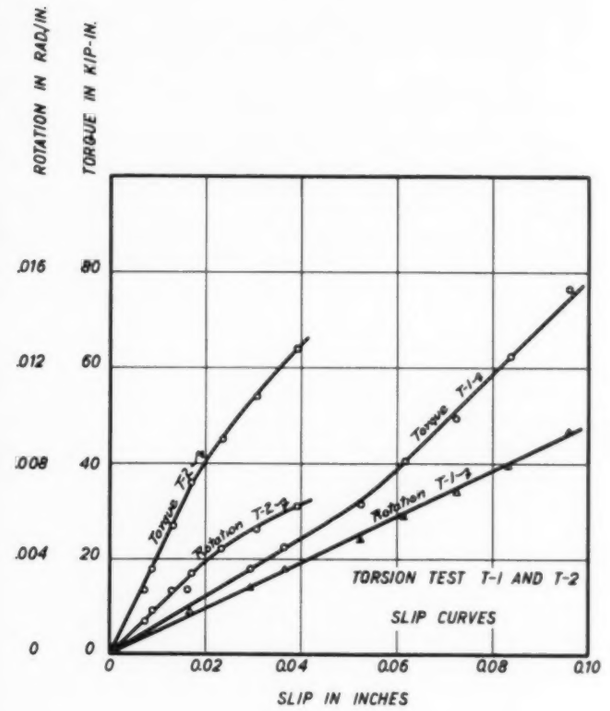
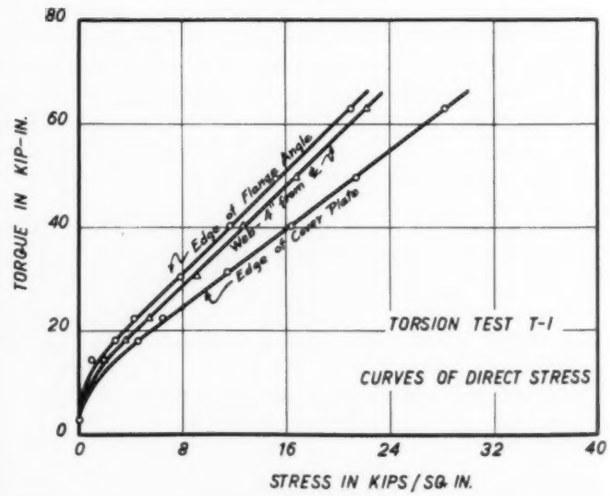


Figure 14

▲

Figure 15



similar to that which Timoshenko uses for the rectangle in torsion (*Strength of Materials*, vol. 1, S. Timoshenko). It should be noted that the stress increases with the square of the twist. Since the built-up beam twists more than the solid beam, these direct stresses are much more important than in the solid beam. The computed stress in the webs checks fairly well with the theory but the stresses in the flange do not check as well. This is probably due to the slipping which occurs, as indicated by the fact that the stresses at the edge of the cover plate and at the edge of the flange angle are quite different, showing a relative slip.

Box Girder Tests—The three box girders were alike except for diaphragm spacing. T-5 had no diaphragms, T-6 had one diaphragm in each end, and T-7 had four diaphragms equally spaced. After T-7 had been tested the first time, the seams were welded with intermittent welds to prevent slippage and the girder retested. This test was called W-T-7.

Very little difference was found in the tests on the three boxes, so apparently the diaphragms were of little effect in the elastic range. These diaphragms were riveted only to the webs as is the usual practice, and thus are relatively ineffective in preventing slip. Since the results for all girders are so very much alike, the test results will be given only for T-6.

In Figure 16 is plotted the twist of the box as the moment was increased. Figure 17 gives curves showing the measured slip and measured stresses in the box.

The measured shears in the flange and web are fairly close to the theoretical. The theoretical shear curve marked 1 is the stress as found from Bredt's formula. The other theoretical curve marked 2 is the sum of the stress from Bredt's theory and the stress due to the twisting of the elements of the box.

The shear curves are computed from the strain rosette data and are the shears acting on the cross-section of the girder perpendicular to the longitudinal axis. The rosettes in the portion of the webs and flanges forming the box showed that the direct stresses on these cross-sectional planes were negligible.

In the rosettes on the outstanding portions of the cover plates and flange angles outside the box section, the measured shear stresses were less than the theoretical shear stresses in the box. The direct stresses determined by these rosettes were appreciable, showing that these sections were not subjected to pure shear.

The direct stress at the edge of the flange as shown in Figure 17 is appreciable for high loads, but in the working range this stress is zero. This stress is a secondary stress and increases with the square of the twist.

The stress in the diaphragm is rather large. This stress was taken on a gauge line 45 degrees with the horizontal since it was thought that the slip in the

Figure 16

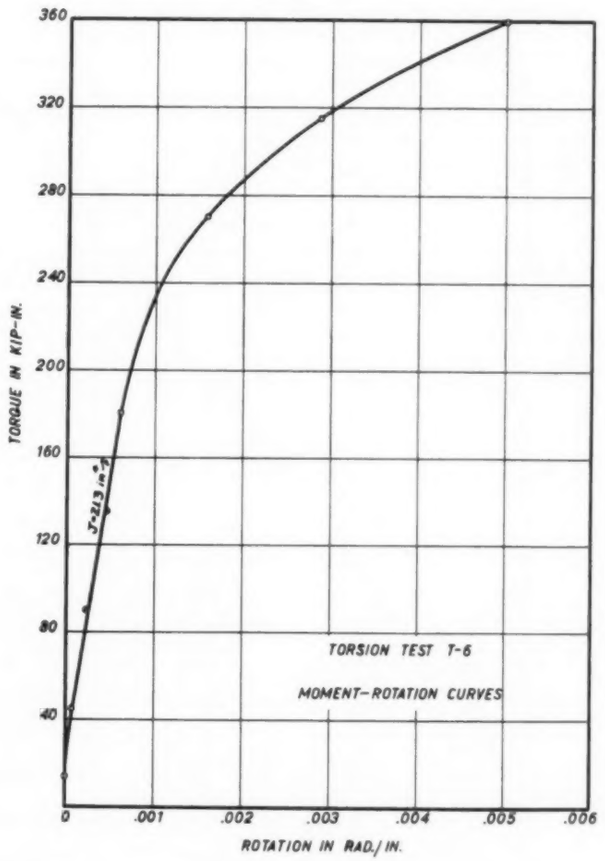
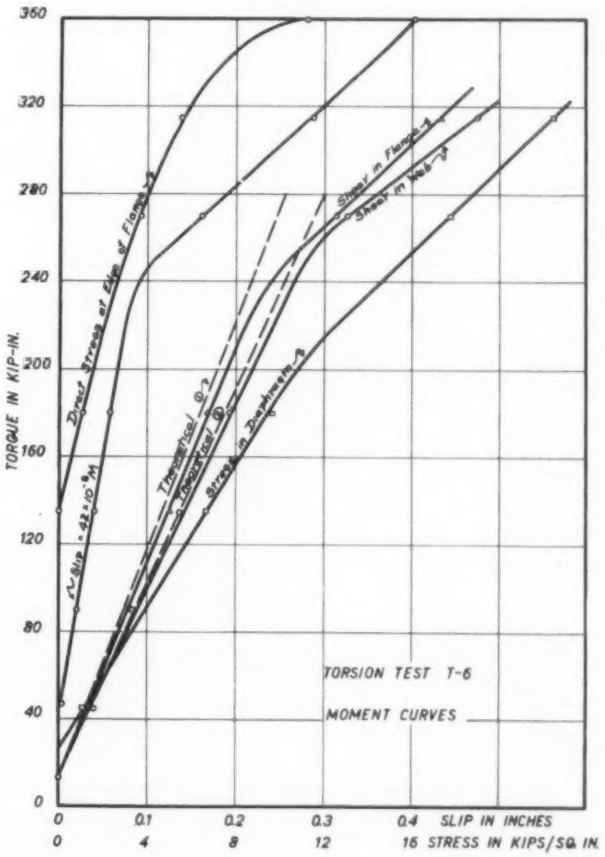


Figure 17



girder would stress the diaphragm with a twisting moment. However, this measured stress is much larger than any which were computed.

The slip in the girder is quite appreciable. The slip did not start with the initial torque but at a torque load of about 45,000 in.-lb. If an initial tension of 32,000 lb. per sq. in. is assumed in the rivets and a coefficient of friction of 0.2 is assumed to be acting between the plates, the load at which slippage will occur is 44,000 in.-lb.

In making these tests, one has to be careful that the torque is applied equally to the whole girder section. The test on T-5 was made with the torque applied to the webs. Figure 18 shows that the top flange and web did not twist equally. The average rotation, however, is about the same as that for T-6 and T-7.

The test results for T-7 after the seams were welded are shown in Figure 19. It is seen that this girder is then much stiffer. The shear curves also check the theoretical computed from Bredt's formulæ, because there was no added twist due to slip.

DISCUSSION OF RESULTS

Pilot Tests—The pilot tests on the box girders showed that the actual twist in a welded box agreed with the theoretical, and that Bredt's formula gives a good value for the torsional constant. These tests also showed that the flange angles, and outstanding edges of the cover plate had little effect on the torsional constant.

The shear stresses on the outside of the box were about 60 per cent greater than the theoretical average value from Bredt's formula. This may be due to the eccentric application of the load through the fillet weld. Since the plate is thin, and the box section very small, this effect is emphasized in this test, and it would be picked up by the comparatively long 2 in. gauge length of the strain gauge. This effect was not noticed in the larger specimens, and it may be a peculiarity of the small specimen.

The pilot tests on the bolted girders probably have no practical significance by themselves. Their chief value is that they exaggerate and emphasize the effects due to slipping in the seams of a riveted girder. As shown by Figure 11, the torsional stiffness of the girder approaches that of the sum of the component parts as the restraints to slipping are removed.

I-Beam Tests—The shear and twist of a long narrow rectangle are given by the formula

$$\tau = \frac{3M}{bc^2}$$

$$\text{and } \Theta = \frac{3M}{bc^3G}$$

$$\text{and } \tau = cG\Theta$$

Figure 18

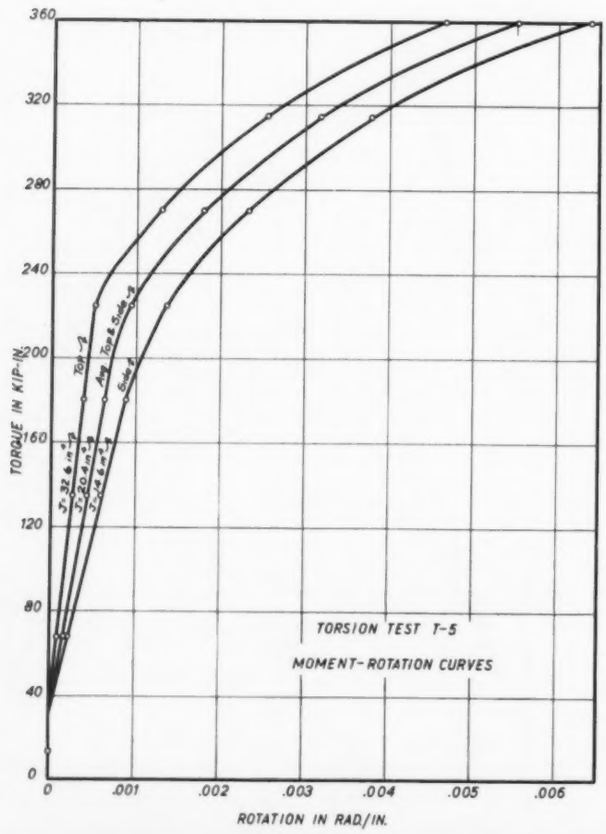
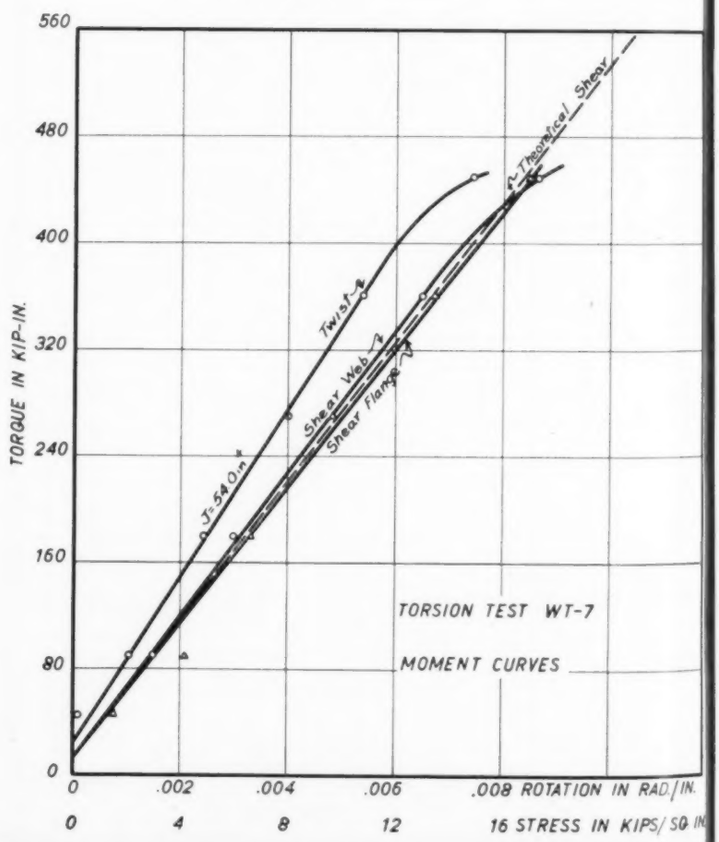


Figure 19



where τ is the shear stress, Θ is the angle of twist in radians per inch, b is the width of the rectangle, and c is the thickness of the rectangle. Since an I-beam is composed of a number of rectangles, an approximate solution for the torsional stresses and twist can be made by dividing the I-beam into a number of rectangles and summing up the sections. However, since the plate thickness enters into the formulæ as square and cube terms, it becomes a question as to what thickness to use when several rectangles are put together. Figure 20a gives a diagrammatic idea of the shear stress distribution when slipping is allowed to occur freely. Figure 20b gives a similar picture for the solid girder in which no slipping occurs. The increased strength due to the greater effective moment arm of the shear stress is apparent. For a riveted girder where some slipping takes place the resultant stress distribution falls between the two distributions shown in Figures 20a and 20b.

In Table III are given the computed torsional constants of the various girders tested. Two values are given, one is computed assuming free slippage and the other is computed on the basis of no slippage. The measured torsional constants for the riveted girders average a little more than one-third of those for the equivalent solid section. The welded girders average 68 per cent. The welded I-beam girders do not have a factor of 100 per cent, since, although the weld does prevent slipping at the toe of the angle, relative strain can occur at the corners of the angles.

For design computations, if a value of the torsion constant of one-third the solid section is used, the computed angles of twist should be fairly close to the actual values. To compute the stresses, these values of Θ should be used in the formulæ. As shown in Table II, the computed stresses are close enough for design purposes. There is little need to go into any refinements in the calculations since the behavior of the girder will depend largely on the slip. Since this slip depends on many factors, and is relatively unpredictable, the behavior of the girders may be quite variable.

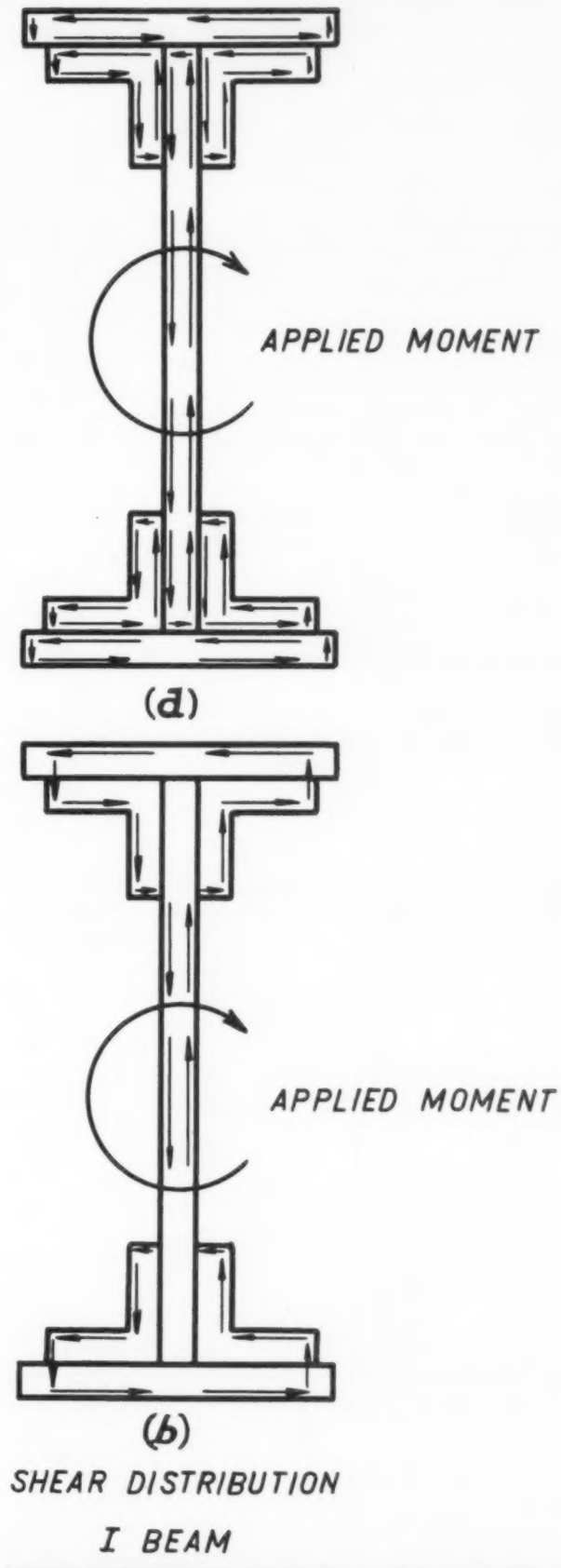
The direct stresses shown in Table II are quite large. Since the direct stresses increase with the square of the twist, and the twist of the riveted girder is three times that of the solid girder, these direct stresses will be nine

TABLE III

Girder	Computed J, in.		Ratio Col. 1 / Col. 2	Measured J, in.^{4}	Ratio Col. 4 / Col. 1	Ratio Col. 4 / Col. 2
	Sum of individual parts	Solid beam				
	1	2	3	4	5	6
T-1	.26	1.20	.22	.49	1.89	.41
T-2	.26	1.20	.22	.83	3.20	.69
T-3	.88	3.64	.24	1.10	1.25	.30
T-4	.88	3.64	.24	2.43	2.76	.67
T-5	.38	57.80	.01	20.40	53.70	.35
T-6	.38	57.80	.01	21.30	56.00	.37
T-7	.38	57.80	.01	20.30	53.50	.35
WT-7*	.38	57.80	.01	54.00	142.00	.94

*Girder T-7 with intermittent welds on seams to prevent slipping.

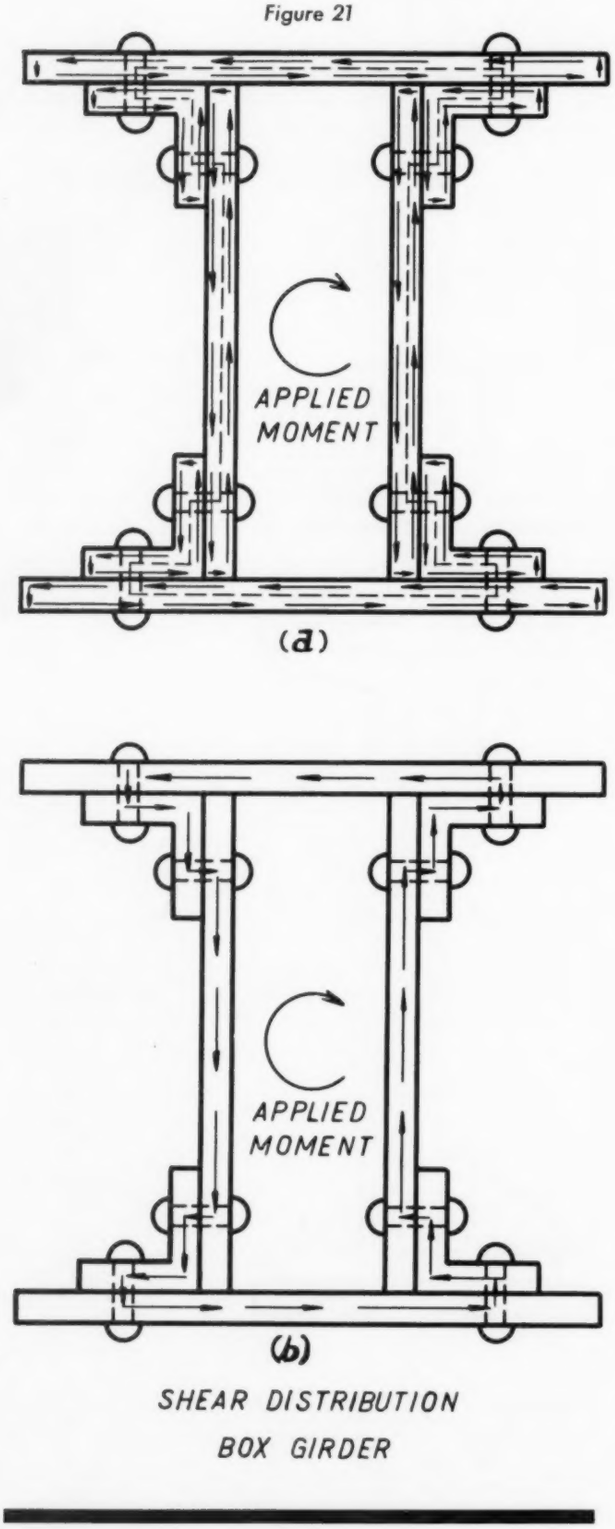
Figure 20



times those in the solid girder. The direct stresses are secondary stresses and therefore have little effect on the ultimate load-carrying capacity of the girder. However, a permanent twist will result in the girder, if these stresses exceed the yield point.

Tests of Box Girders—The tests on the riveted girders showed that they behaved about as predicted by Bredt's formula, except as this is modified by slip. Table II shows that the torsion constant is about one-third that which is predicted by Bredt's theory. In computing this torsion constant, one must remember that the built-up rectangular section is not a box but a figure more like an H-beam since the stress in the flange must travel outside the box section to the rivet and back again before it is transferred to the web. Figure 21a shows the section acting in the stress transfer. The dotted line shows the edges of the section used to compute A in Bredt's formula. Figure 21a also gives a diagrammatic picture of the shear stress transfer in a box section where slipping is allowed to occur. Figure 21b shows the shear stress transfer when no slipping occurs. The reason for the effectiveness of the box section is obvious when one notices the large effective moment arm of the shear stress.

Figure 21



SHEAR DISTRIBUTION
BOX GIRDER

The shear curves for Figure 17 show that Bredt's theory gives a fair approximation for the average shear stress, and this computation can be made more exact by adding the stress due to the twist of the sides of the box. Bredt's theory assumes that the shear stress is constant through the thickness of the box, and when the thickness of a wall is small in comparison with the distance of the plate to the center of gravity of the box, this approximation is small. Actually, the stress increases from the inside of the box wall to the outside. The correction added to the average shear as given by Bredt's formula gives the maximum stress on the outside of the box. Since the riveted box twists more than the solid box, the correction is larger for the riveted box. In Figure 19 this correction is negligible since the welded box did not slip resulting in a smaller angle of rotation.

The direct secondary stress in the box as shown by Figure 17 can be safely neglected in the working range.

The stress in the diaphragms was larger than expected. It may have been affected by local bending and readjustments. The direct stresses for the welded girder were much less than those for the riveted girder.

The measured slip in the joints as given in the curves is the total of the slip in all the seams. The actual slip in any one corner of the box is the plotted slip divided by four. It can be shown (N.A.C.A. Report No. 502, "A Theory for Primary Failure of Straight Centrally-loaded Columns") for a box with free slipping seams (no rivets or welds) that the longitudinal motion or warping of any point in the box section is the product of the x and y coördinates of the point multiplied by the angle of twist Θ when slipping occurs in all the seams. Since in the same corner, the web and the flange warp in opposite directions, the relative slip is equal to twice this product. Thus the relative slip in the corner of a box of height h and width w would be $wh\Theta/2$. When the box is riveted, the actual slip is less than the computed value above, and this actual slip is a measure of the torsional constant. For the gauge lines on which slip was measured in the test, the relative slip for loose seams is 42Θ. For girder T-6 the measured slip for the straight line portions of the slip curve is $10.5M(10)^{-8}$ for one seam.

Then the twist angle is given by:

$$\Theta = \frac{M}{GJ \text{ (no slip)}} + \Theta \text{ (due to slip)}$$

or $$\Theta = \frac{M}{GJ} + \frac{10.5M(10)^{-8}G}{42G}$$

$$= \frac{M}{G}\left(\frac{1}{57.8} + \frac{1}{33.3}\right) = \frac{M}{G}\left(\frac{1}{21.2}\right)$$

The value of 21.2 is the effective torsion constant, and provides a check on the value computed directly from the curve. This check is not absolutely independent, since the value of Θ is inherent in the second term.

CONCLUSIONS

The torsion tests herein reported indicate that the following results may be concluded.

1. Fabricated I-beam and box sections do not behave the same as the equivalent solid sections.

2. The slip which occurs in the seams of built-up sections is appreciable under torsional loading and increases the twist and stresses of such sections markedly.

3. The torsional constant of the riveted built-up I and box sections tested in this program was about one-third that of the equivalent solid section.

4. If the twist angle Θ is computed using the reduced value of the torsional constant, shear stresses computed from this value will agree fairly well with the actual values for I-beams.

5. Direct secondary stresses are quite large in the built-up I section and may be important.

6. Direct secondary stresses are small in a box section when stressed in the working range.

7. The direct stresses vary with the square of the angle of twist.

8. Bredt's theory gives good approximate values for the shear stress in box sections.

9. In applying Bredt's theory to a fabricated box section, one must be careful to use the section bounded by the stress path. This will not be a rectangular section when the corners are formed by riveted angles.

10. Built-up welded I-beams fabricated with flange angles had a torsion constant two-thirds that of the equivalent solid section.

11. Built-up welded box girders had a torsional constant close to that predicted by Bredt's theory.

12. The outstanding legs and parts of a fabricated box section not included in the box have little effect on the torsional properties of the section.

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5. Box Girder Buckling Tests

▲ THIS REPORT describes the results of three series of buckling tests on a number of box girders.

The first series of tests investigated the length-width of box or l/b ratio of the girders. This series was divided into two parts. In the first part, the girders were loaded with a vertical load only, and in the second part the girders were loaded with both a vertical load and a lateral load applied to the top flange. The lateral load caused a lateral moment which was 10 per cent of the vertical moment. Both riveted and welded girders were tested with l/b ratios varying from 110 to 60. In no test was there any sign of lateral buckling, and all the girders failed because of local yielding.

The second series investigated the coverplate width-thickness or w/t ratio of the girders. This series was also divided into two parts. In the first part, the girders were loaded on the top flange at diaphragm points, and in the second part, the girders were loaded through a rail at a point half-way between the diaphragm points. Welded girders were tested with w/t ratios varying from 32 to 96. The ultimate strength of the girders and the buckling strengths are given in this report.

The third series investigated the web height-thickness or h/t ratio of the girders. Welded girders were tested with h/t ratios varying from 320 to 144. The girders with the h/t ratio of 320 had a horizontal stiffener. The buckling loads of the web plates were computed by Southwell's method and the loads at which buckles could first be noticed are given in this report.

In this investigation, tests were conducted to determine the buckling strength of the component parts of box girders, and the buckling strength of the girders as a whole. Since the present tendency of design is to use higher allowable design stresses, a more exact knowledge of buckling loads is necessary. This is particularly true when high-tensile steel is to be used. The buckling strength of a structure in the elastic range does not depend on the strength of the steel but on the form and the modulus of elasticity. A more precise knowledge of buckling is therefore necessary when high-tensile steel is used in order to take advantage of the higher strength of such steel.

The allowable thickness ratios in many box girder specifications have been taken from bridge specifications such as the A.R.E.A. Specifications for Steel Railway Bridges. These bridge specifications were primarily written for I-beams and do not necessarily apply to box girders. Since the edges of the plates forming box girders are usually held more rigidly than the edges of the plates in I-beams the plates forming the box girder should not buckle as easily.

In addition, the maximum allowable compression stresses in a box girder are usually much too low. They have been taken from old I-beam formulæ which do not apply. The beam in a bending test must twist if lateral buckling due to bending occurs. Such buckling may occur rather easily in I-beams which are relatively weak in torsion. Box crane girders of normal proportions are

very rigid in torsion and this type of buckling can not occur.

Three series of girder tests were made in this investigation.

In Table A is given a summary of the pertinent data of the tests and test girders. The following is a more extended description of these tests.

l/b Ratio Tests—The first series of tests investigated the unsupported length-width ratios of the girders. Welded girders G-1, G-2, G-3, G-4 with *l/b* ratios of 80, 60, 40, and 30, and riveted girders G-15 and G-16 with *l/b* ratios of 58.2 and 29.1 respectively, were subjected to bending tests. These girders were loaded symmetrically about their centerline by two vertical loads placed at distances 5 ft. from each end. The *l/b* ratios given above are the ratios for the unsupported spans between the load points. The total overall span-width ratios were 110, 90, 70, and 60 for the welded girders and 78 and 61 for the riveted girders.

A high-tensile steel girder G-17 was included in the above series. This girder is a duplicate of G-4 except that it was made of Mayari-R steel.

In Figure 1 are shown the essential details of the girders tested in the *l/b* series. These girders were supported on rollers at each end and were loaded through rollers located 5 ft. from each end. Figure 2 is a picture of G-2 in the testing machine. Since the girders were loaded through rollers, there was very little restraint offered the test beam at the load points.

In order to simulate design practice with regard to lateral load allowances, four of the above girders were tested with a lateral load applied to the top flange at the span center in addition to the vertical load. The lateral load was of such a magnitude that the lateral

TABLE A

Series; purpose of test	Girders tested	Variable: Clear span	Variable: Overall span	How tested description	Material	Measurements
		l/b ratio				
1. Length-width ratios	G- 1	80	110		Welded	Stresses:
Initial tests	G- 2	60	90		Welded	10 in. W. S. G.
	G- 3	40	70	Symmetrical vertical	Welded, Carbon steel	Vertical deflection:
	G- 4	30	60	loads placed at dis-	Welded	Strings and mirror
	G-15	58	78	tances 5 ft. from ends.	Riveted	Lateral deflection:
	G-16	29	61		Riveted	0.001 Federal dial
	G-17	30	60		*	
	G- 2a	60	90	Vertical load plus lat-		
Retests	G- 3a	40	70	eral load applied to		
	G- 4a	30	60	top flange.		
	G-17a	30	60			
		w/t ratio				
2. Spacing of webs	G- 5	32		Girders 18 ft. long.	Welded	Stresses:
Width-thickness ra-	G- 6	48		Loaded with sym-	Welded	10 in. W. S. G.
tios of cover plates	G- 7	64		metrical vertical	Welded, Carbon	Deflections:
Initial tests	G- 8	80		loads on diaphragms	Welded steel	Vertical—Ames dials
	G- 9	96		at third-points.	Welded	Lateral—Ames dials
	G-18	64			*	
	G- 5a			Girders loaded		
	G- 6a			through rail by verti-		
Retests	G- 7a			cal loads at points		
	G- 8a			half-way between		
	G-18a			diaphragms.		
		h/t ratio				
3. Height-thickness ra-	G-10	320		Girders are 18 ft. long.	Welded	Stresses:
tio of web	G-11	192		All girders loaded by	Welded	10 in. W. S. G.
	G-12	176		two vertical loads at	Welded, Carbon	Deflections:
	G-13	160		one-third points.	Welded steel	Vertical—Ames dials
	G-14	144			Welded	Lateral—Ames dials

*Welded Mayari-R steel

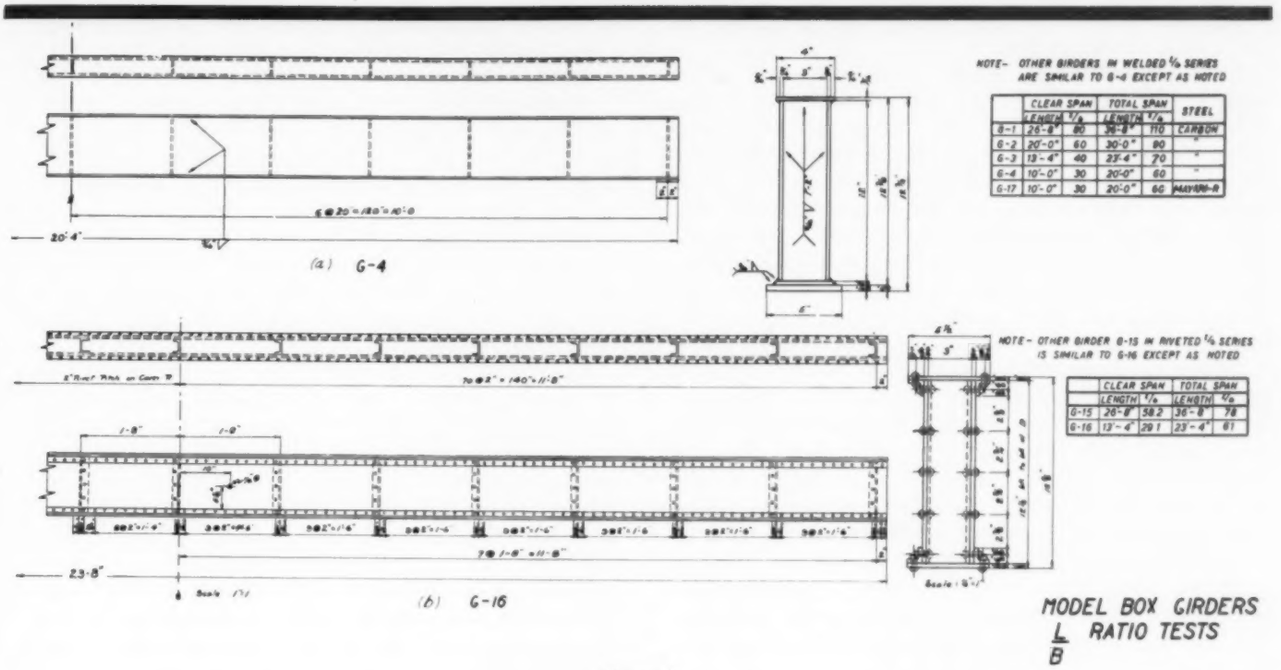

NOTE- OTHER GIRDERS IN WELDED ½ SERIES ARE SIMILAR TO G-4 EXCEPT AS NOTED
NOTE- OTHER GIRDER G-15 IN RIVETED ½ SERIES IS SIMILAR TO G-16 EXCEPT AS NOTED
(a) G-4
(b) G-16
MODEL BOX GIRDERS
L RATIO TESTS
B

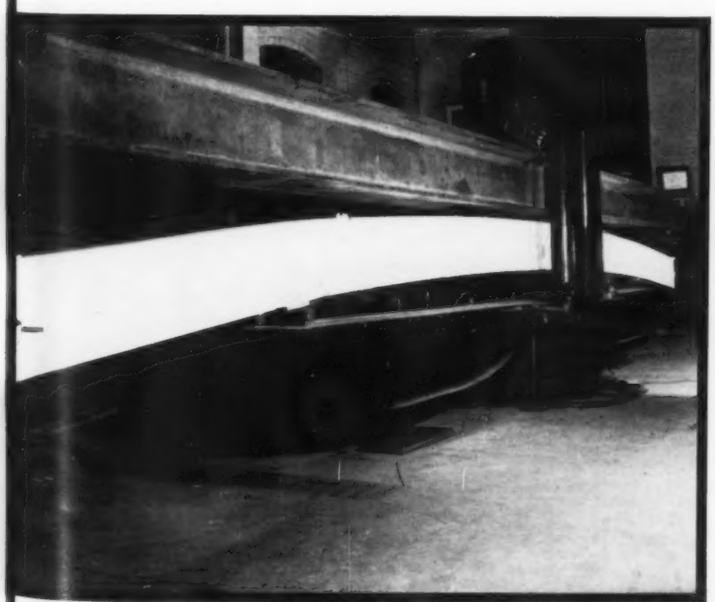
Figure 1

moment at the span center was 10 per cent of the vertical moment. The retests were made on girders G-4, G-3, G-2, and G-17, and the retest specimens are denoted in this report with the letter "a" following the girder number, as for example G-4a. Figure 3 shows a picture of girder G-4a in the testing machine.

Figure 3 →

▲

Figure 2



In the first tests, the girders had been loaded to their ultimate load and failure occurred with excessive deflection and local buckling. These girders had to be restraightened for the retests. This was done by turning the girders upside down in the testing machine and applying load until the girders were straight and the buckles had disappeared. The first loading and the straightening process probably affected the yield point of the steel but should have had little effect on the other

properties. The retests were made with the girders in the upside down position.

The stresses in the girders were measured on a number of gauge lines at the span center with a 10 in. Whittemore strain gauge. The vertical deflection was measured on both sides of the girder by the stretched wire and scale method. The lateral deflections of both the top and bottom flanges of the girder were measured with a 0.001 in. Federal dial attached to a portable arm 10 ft. long. The lateral load on the girders was applied by tightening a nut on the lateral load rod, and the magnitude of the lateral load was determined from the strain in the rod.

w/t Ratio Tests—The second series of tests investigated the spacing of the webs, and the width-thickness ratio of the cover plate. These girders were 18 ft. long and loaded with symmetrical vertical loads on the diaphragms at the third-points. The girders in this series were G-5, G-6, G-7, G-8, G-9, with width-thickness ratios for the coverplate of 32, 48, 64, 80, and 96 respectively. Girder G-9 had a longitudinal stiffener along the center line of the coverplate. A high-tensile girder G-18 made of Mayari-R steel, was also tested in this series. This girder is a duplicate of G-7. Figure 4a shows the details of the girders in this series.

Retests were made on four girders in this series identified as G-5a, G-6a, G-7a, and G-8a. Since the girders are loaded through a trolley rail in actual practice, the retests were made to determine whether the coverplate was appreciably weaker when loaded through a rail at a point half-way between the diaphragms. The rail would have a tendency to stiffen the coverplate which would strengthen the coverplate, whereas the wheel load half-way between the diaphragms would stress the coverplate and thus weaken it.

To simulate the rail a 3 in., 5.7 lb. I-section was used in these retests. Girder G-5a was loaded with a single concentrated load at the span center which point is half-way between two diaphragms. Girders G-6a, G-7a, G-8a, and G-18a were loaded through the rail at two points, each 2 ft. from the span center. These load points are also half-way between diaphragms.

The girders had been fabricated with the diaphragms welded to the top flange and webs, thus leaving a space between the diaphragm and bottom flange. Therefore in the retests, the diaphragms probably offered no support to the rail, since the girders were upside down in the retests and the load was applied to what had been the bottom flange.

The deflections of the girders in this series were measured with Ames dials. The stresses were measured with a 10 in. Whittemore strain gauge. The local bending and buckling of the top coverplate was measured with a small contact dial measuring to 0.001 in. over a grid of very small squares for a distance of 12 in. on each side of the center line.

h/t Ratio Tests—In the third series of tests, the height-thickness ratio of the web was investigated. The girders in this series were G-10, G-11, G-12, G-13, and G-14. These girders had web height-thickness ratios of 320, 192, 176, 160 and 144 respectively. The girder with the height-thickness ratio of 320 had a longitudinal stiffener the whole length of the girder at a distance about one-fourth of the way down from the compression flange. Figure 4b shows the details of these girders. These girders were loaded at the third-points so that there was pure bending moment with no shear in the center third of the beam. There were no diaphragms in the center third of the girder. In the sections of the beam between the load point and the end support, there were vertical diaphragms and also a longitudinal stiffener along the center line of the web to prevent shear buck-

Figure 4

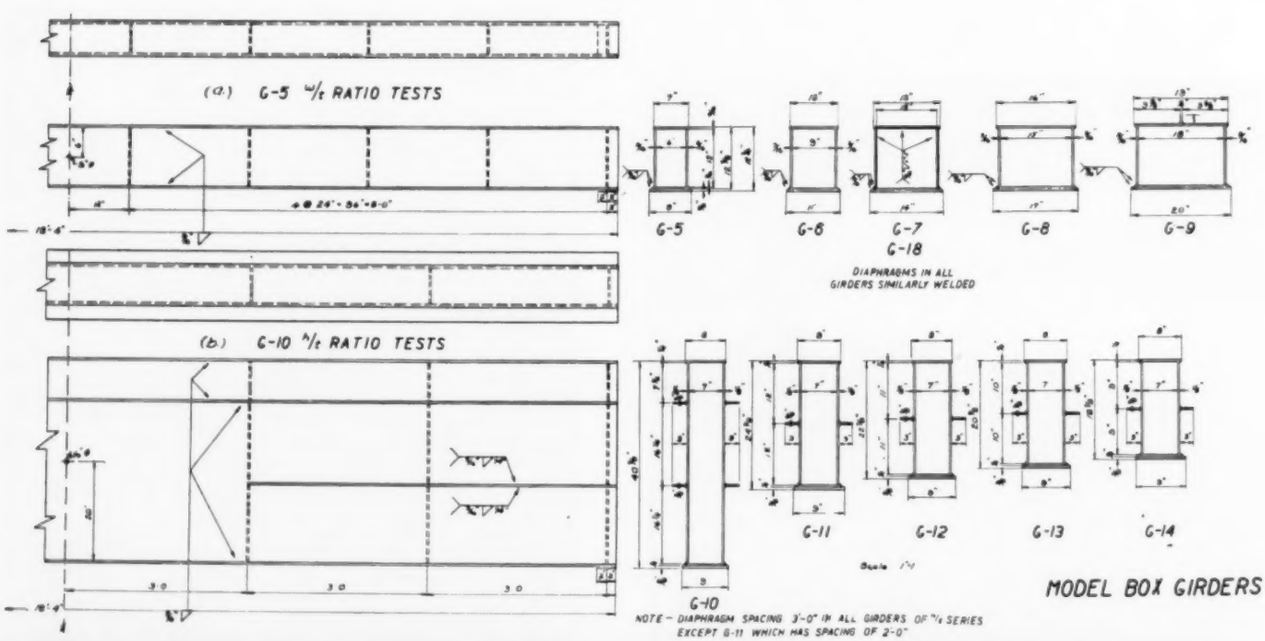


TABLE I

l/b Ratio Tests—Vertical Load Only

1	2	3	4	5	6	7	8	9	10	11	12	13
Girder	Theoretical buckling moment, kip-in.	Maximum vertical load, kips	Maximum applied moment, kip-in.	Net vertical moment of inertia, in.^{4}	Vertical section modulus, in.^{3}	Maximum computed stress, M/S, kips per sq. in.	Maximum computed stress, parabolic distribution, kips per sq. in.	Average yield point of flange material, kips per sq. in.	Unsupported l/b	Allowable stress formula, kips per sq. in.	Total l/b	Allowable stress formula, kips per sq. in.
G- 1	2700	35.0	1050	112.9	18.2	57.8	48.2	44.9	80.0	0	110	0
G- 2	3300	31.5	945	114.0	18.4	51.3	42.7	42.4	60.0	7.4	90	0
G- 3	4240	33.3	1000	114.0	18.4	54.4	45.2	44.7	40.0	13.3	70	3.6
G- 4	4970	32.3	965	117.5	18.9	51.1	42.6	42.7	30.0	15.4	60	7.4
G-15	2480	11.3	1240	151.5	22.0 26.3	56.3 47.3	42.8	42.0	58.2	4.8	78	0
G-16	2860	40.1	1200	147.8	21.4 25.6	56.0 47.0	42.8	45.9	29.1	14.7	61	8.0
G-17	4970	46.0	1380	117.0	18.9	73.1	60.8	63.2	30.0	21.3	60	4.0

ling and to assure failure in the center third of the beam.

Girder G-10 was tested twice. It was first loaded to 169,000 lb. at which load the loading beam started to fail. Then, most of the longitudinal stiffener was burned off until it extended only about an inch from the web. The girder was retested and then failed at a load of 118,000 lb.

The stresses in these tests were measured in the girder with a 10 in. Whittemore strain gauge. The vertical deflections of the girder were measured with Ames dials and the local lateral buckling of the web was also measured with Ames dials. The location of the points where the lateral deflection was measured can be seen in the figures showing the lateral deflection of the girders.

The stresses in the diaphragms were measured in many of these tests with strain rosettes. The stresses on the inside of the plates of the box girders were also measured. These strain rosettes were read on an electric gauge.

Remarks on Buckling Theory—In determining buckling strength, it is sometimes difficult to define the buckling load, since buckling is primarily a mathematical concept based on assumptions which are often different from actual conditions. The buckling of a column is defined as the load which causes excessive lateral deflection, when the column is perfectly straight, the load perfectly centered, and all strains perfectly elastic. For long slender columns, these conditions can be approximated by test and the test loads will be fairly close to the theoretical. Theoretically there is no lateral deflection until the buckling load is reached. Actually in practice, no column is perfectly straight and some lateral deflection occurs even at low loads. This lateral deflection increases more rapidly than the load, and the column may approach but never attain its true buckling value.

The same criterion applies to the buckling of plates. The problem is here more difficult than the problem of the column, since a plate in a structure is relatively more crooked than a column. In addition, in many structures such as box girders, the remaining portion of the structure restrains the component plates from buckling. When the web buckles, for example, the flanges prevent excessive lateral deflection, and thus the webs will hold the buckling load while additional load applied to the beam is taken by the flanges. Similarly, if the flange buckles, the box girder acts as a U-beam and the webs take the additional load until they too buckle. Thus normally, the box girder as usually designed, will not fail until both the web and flanges buckle. The load at which this occurs will depend upon the proportions of the webs and flanges. Since the plates in the girder are not perfectly straight, some local lateral deflection will occur at very low loads.

There is a method, originally proposed by Southwell (*Proceedings Royal Society*, Vol. 135, 1932, p. 601; N.A.C.A. Tech. Notes 658, E. E. Lundquist), by which the theoretical buckling load can be computed from the lateral deflection readings. This method was applied to finding the buckling loads in the l/b series and the h/t series but was not applicable to the w/t series due to the variable eccentricity on the plate caused by the deflection of the beam.

In all these tests, an attempt was made to design the girders so that the failure would be influenced primarily by the variable being investigated. In other words, in the girders with the variable h/t ratio, the flanges were made thick enough so that failure would not occur in the top flange. However, in many of the girders, the yield point was reached before the buckling load, and then the girders failed as a unit.

TEST RESULTS

l/b Ratio Tests—Table I gives the test results for this series. With the exception of G-15 all girders were loaded symmetrically with two vertical loads spaced 5 ft. from each end. G-15 was loaded with a center load since the supporting beam was not strong enough for the other type of loading. As can be seen in the table, all the welded carbon steel girders failed at about the same load. Both riveted girders failed under the same bending moment. The alloy steel girder failed at a load

TABLE II

l/b Ratio Tests—Vertical Load Plus Lateral Load

1	2	3	4	5	6	7	8	9	10
Girder	Maximum applied vertical moment, kip-in.	Vertical section modulus, in.^{3}	Maximum applied lateral moment, kip-in.	Lateral section modulus, in.^{3}	Maximum computed vertical stress = M/S, kips per sq. in.	Maximum computed lateral stress = M/S, kips per sq. in.	Total maximum computed stress, kips per sq. in.	Total computed stress parabolic distribution, kips per sq. in.	Average yield point flange, kips per sq. in.
G- 2a	885	18.4	82.0	6.1	48.2	13.4	61.6	51.2	42.4
G- 3a	940	18.4	94.5	6.1	51.2	15.5	66.7	55.5	44.7
G- 4a	935	18.9	94.5	6.2	49.6	15.2	64.8	54.0	42.7
G-17a	1285	18.9	122.0	6.2	68.0	19.7	87.7	73.0	63.2

which was in proportion to the yield point. Evidently the length of span did not influence the strengths of the girders tested.

Column 7 gives the stresses in the girder based on the common beam formula. Since the computed stresses are higher than the yield point of the material, these stresses are not true stresses. Column 9 gives the yield points of the steel as determined from tensile coupons. This yield point was about 42,000 to 45,000 lbs. per sq. in. and therefore the stress distribution of the beams is not linear nor does the beam formula apply above this stress. If the resisting load in the box is assumed to vary parabolically, the stresses will be five-sixths that of the ordinary beam formula. Column 8 gives these stresses, and they are seen to check very well with the yield point of the material as determined from the tension test. This is evident when comparing the values of column 8 with column 9.

It should be pointed out that the assumption of parabolic distribution for these tests is an empirical one and may not apply to girders of different cross-sectional proportions. The important fact is that the girders carried greater moment than that which would be predicted by using the yield-point stress of the material in the ordinary beam formula.

Column 13 gives the allowable stress in the compression flange of the girders using the total span in the common specification formula:

$$fc = 18{,}000 - (0.35)\left(\frac{l}{r}\right)^2$$

These stresses are much less than those present at the ultimate load. The computed allowable stresses by this formula are appreciably higher than those given by most other specification formulæ. Column 11 gives the stresses computed by this formula on the basis of the unsupported span. These stresses are higher than those given in column 13, but in most crane design the total span is used in this computation.

In column 2 is given the theoretical buckling moment of each girder as computed by the stability formulæ derived by Timoshenko (*Theory of Elastic Stability*, Stephen Timoshenko, p. 242; N.A.C.A. Technical Note No. 601, "The Lateral Instability of Deep Rectangular Beams," C. Dumont and H. N. Hill). With the exception of G-15, these moments were computed assuming that a constant moment acted along the whole girder. This assumption will give buckling moments which are smaller than the actual buckling moments. For the longer girders the approximation is small. The formula, by which these moments are computed is:

$$M_{cr} = \frac{\pi\sqrt{B_1 C}}{l}$$

M is the buckling moment in inch-pounds.
l is the span length in inches.
B_1 is the lateral rigidity and equals the lateral moment of inertia I_2 times the modulus of elasticity E,
C is the torsional rigidity and equals J the torsional constant times G the modulus of shear.

The buckling moment of G-15 was computed from the formula:

$$P_{crit} = \frac{16.93\sqrt{B_1 C}}{l^2}$$

where P_{crit} is the buckling load. This last formula applies to a beam loaded at the center, and thus should give close results for a crane when the trolley is at the center.

These formulæ give buckling loads much higher than those given by most design formulæ, because the above formulæ take into account the torsional strength of the girders. Since box girders are so strong in torsion, neglecting this factor seriously underestimates the strength of such girders. In I-beams where torsional strength is relatively small, this neglect does not result in such a large error.

In computing the strength of the riveted girders, the value of the torsion constant was multiplied by one-third to take into account the effect of rivet slip (*Torsional Properties of Fabricated I-Beam and Box Sections*, I. E. Madsen).

These buckling formulæ apply only in the elastic range. Since the theoretical buckling loads corresponded to stresses far above the yield point, the girders failed due to yielding of the steel.

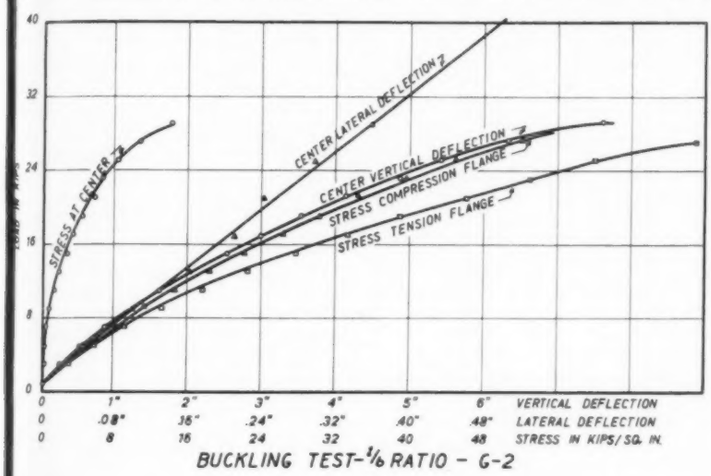
The ultimate loads of the girders with the 10 per cent lateral moment are given in Table II. These ultimate loads are almost as large as those for the tests in which no lateral load was present. If the lateral stress is added to the vertical stress, the maximum stresses in these tests are greater than those in the vertical load tests. However, since these computed stresses are above the yield point, the formulæ do not

apply. The stresses based on parabolic distribution of the resisting bending force are larger than the yield point as shown in Table II. It is uncertain, however, that these yield-point values apply, after the cold-working the girders were subjected to in the straightening process.

In this report, plotted curves of test results will be reported for the tests on girder G-2 and G-2a. The results for the other girders in this series are very similar.

Figure 5 gives a number of curves which show the behavior of this girder under the vertical test load. Curves for the center vertical and lateral deflection, and the stresses in the compression flange, tension flange, and at the center line of the girder are given in this figure.

Up to a load of about eight kips, the stresses and deflection were linear and checked the theoretical.

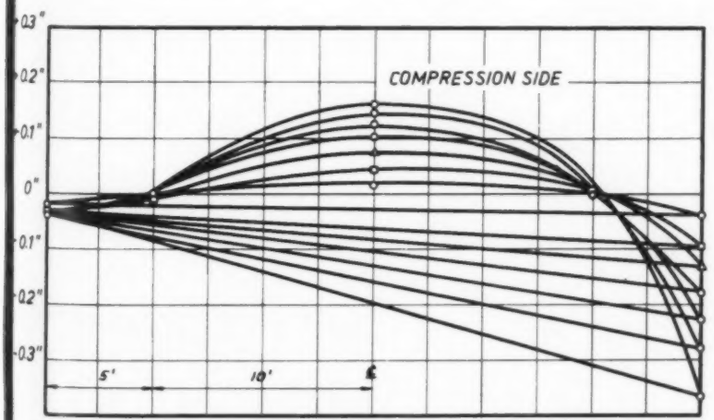


BUCKLING TEST-1/6 RATIO - G-2

Figure 5

▲

Figure 6



BUCKLING TEST-1/6 RATIO

LATERAL DEFLECTION - G-2

Above this load, the stresses and deflection increased somewhat more rapidly than the load. It should be noticed that the compression stress above this load is less than the tension stress and the difference becomes relatively greater. This is rather unusual, for the stresses in both the tension and compression flanges should have been the same, since the girder section was symmetrical and had no holes.

The effect is believed due to internal welding stresses. After the flanges are welded to the webs, there is an initial yield-point tension in the weld metal and the material next to it. Then as the load is applied to the beam, the material on the tension flange which is at the yield point refuses to take more stress. The weld metal on the compression flange, which has yield-point tension, takes compression however, and the resultant stresses are the same as those present in an unsymmetrical beam with a portion of the tension flange missing. The resultant stress distribution is similar to that in a riveted girder.

As the tension increases in the tension flange, the material adjacent to the welds, which is close to the yield point, becomes ineffective, resulting in less effective flange area, and moving the apparent neutral axis still closer to the compression flange. This is shown in Figure 5 by the curve showing the stress at the center of the girder. Since the initial tension is an internal stress, there must be an initial compression to balance this tension. The internal compression stress is spread out on a much larger area than the tension stress and consequently the compression stress is much lower than the tension stress, so this area does not go out of action at the lower loads. If the girders are stressed above the yield point, the internal stresses are largely relieved, and this will be shown in the curves for G-2a where the measured values of the tension and compression stresses were practically alike.

This effect of the shifting of the neutral axis was noticed in all the girders, though in no case was the effect larger than in G-2, and in most cases, it was smaller. The effect was usually greater than the net section effect in the riveted girders.

One curve which should be noted is the curve for the center lateral deflection. This curve is a straight line. If there were any tendency for lateral buckling, this curve would show increasing deflections for the same load increments, as the load was increased.

The measured lateral deflection is due to the fact that the girders were fabricated with the usual structural tolerances, and the top and bottom flanges were not perfectly parallel. The measured lateral deflection here corresponds to a lateral load of 0.01 the vertical load.

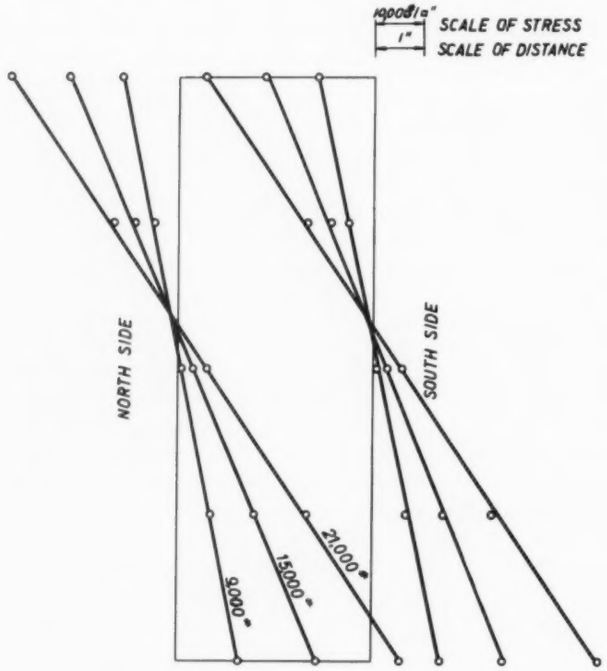
The lateral deflections, as measured on the compression flange, are shown in Figure 6. The measured lateral deflections on the tension flange were practically the same. This shows that no twist occurred in the test. The lateral deflection plotted in Figure 5 is the average of the tension and compression measurements.

In Figure 7, the stress distribution at the center of the girder is shown at three loads. The neutral axis moved towards the compression flange as the load was increased. The strain gauge measurements showed that the straight line strain distribution remained linear up to the ultimate load. This was true in all the girders

except those of the h/t series in which the webs buckled.

Figure 8 shows curves for the deflection and the stresses in girder G-2a. The stresses are the average stresses in the flange. Since there was a lateral load, the stresses varied from one flange edge to the other. The stress values in both the tension and compression flanges are about equal as can be seen in the figure. This shows that the internal stress condition shown by the test on G-2 had been relieved. The stresses and also the deflection are linear up through the proportional limit.

The lateral deflection curve at the center is plotted against the lateral moment. The measured lateral deflection is practically as large as the vertical deflection,



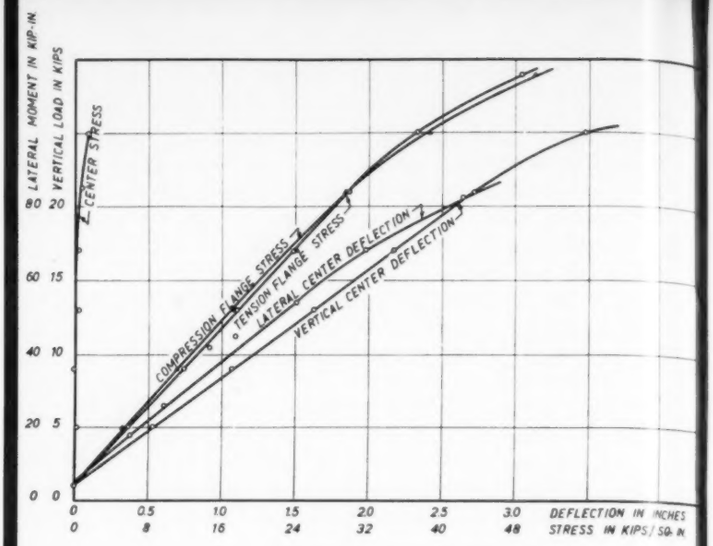
BUCKLING TEST-1/6 RATIO
G-2-STRESS DISTRIBUTION

Figure 7

and is the average of the readings on the tension and compression flanges. The deflection on the compression flange was about 5 per cent greater than that on the bottom flange showing that the lateral load, which was applied to the top flange, caused some twist.

In Figure 9, are shown the curves of the lateral deflection of the top flange, as they were actually measured. The points where the curves cross the zero line are the points where the vertical load was applied.

w/t Ratio Tests—The results of the w/t test series are summarized in Table III. Column 2 gives the maximum loads and column 3 gives the maximum moments applied to the girders when failure occurred. The computed maximum stress based on the common beam



BUCKLING TEST-1/6 RATIO
G-2A

Figure 8

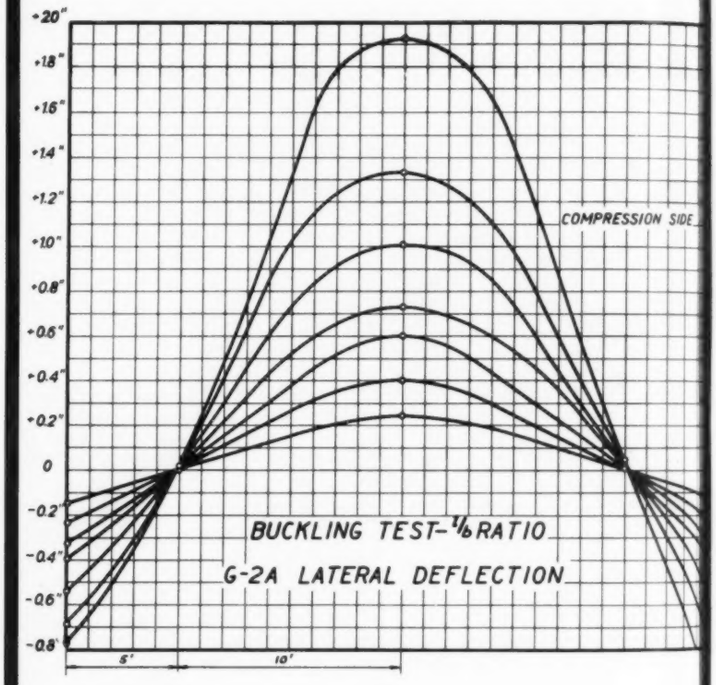
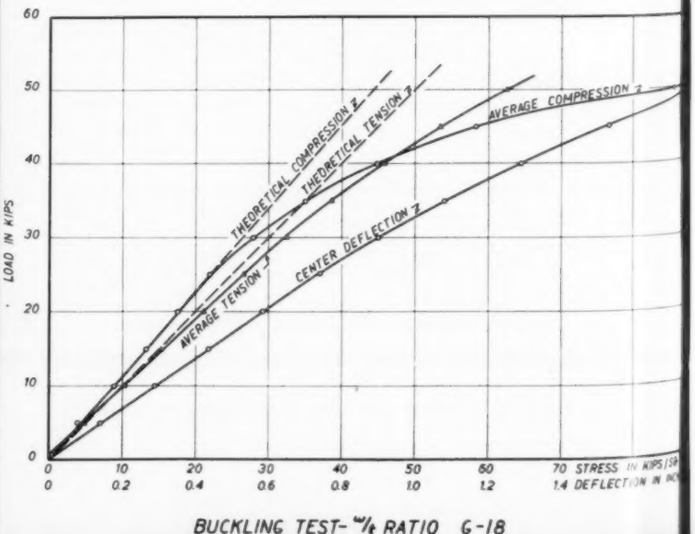


Figure 9

▲

Figure 10



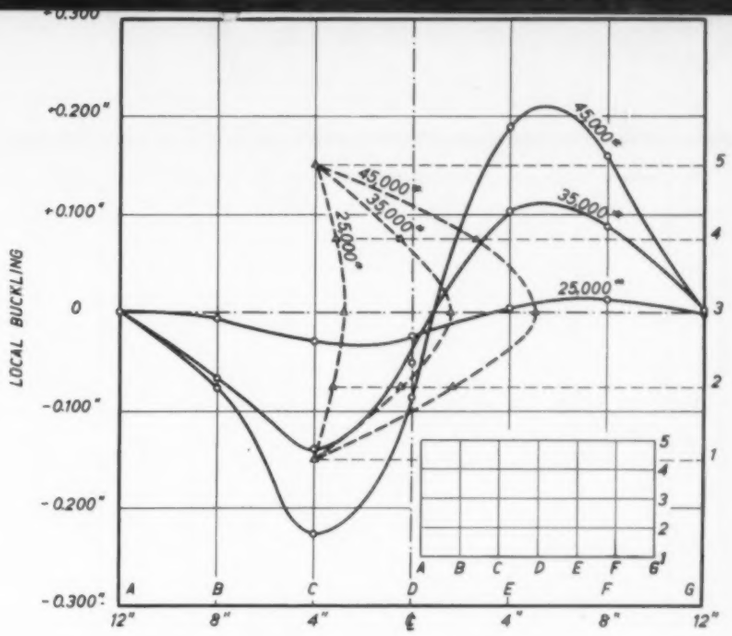
BUCKLING TEST-1/4 RATIO G-1B

theory is given in column 6. For the girders without the rail, the maximum computed stress is less than the yield point with the exception of girder G-5. This shows that buckling must have contributed to the failure of the girders. The buckling of the cover plate in most of the girders tested in this series may have occurred long before the girder failed as a whole.

Column 9 gives the theoretical buckling stress of the girders as developed by Timoshenko assuming the edges of the plates to be simply supported. The measured stresses in the flange plates when they buckled are given in columns 10 and 11. These stresses are appreciably smaller than the theoretical.

As stated in the introduction, the buckling loads for the girders in this series are difficult to determine. Two criteria were used to determine the buckling stresses. The buckling loads given in column 10 were determined from stress curves, similar to those in Figure 10 for girder G-18. In this figure which is typical of this series, the compression stress is less than the tension stress at the lower loads. This is due to the initial tension stress in the girder caused by welding stresses discussed in the previous section. At the higher loads, the compression stresses increase more rapidly than the tension stresses, and become greater than the tension stresses at a load of 40,000 lb. for girder G-18. The load at which the compression stress begins to increase more rapidly than the tension stress is about 26,000 lb. and this was taken as the buckling load of the compression flange. This load was taken as the buckling load because at this load, the compression flange ceases to be fully effective (due to buckling) and the neutral axis begins to approach the tension flange. The stress in the compression flange then increases more rapidly than that in the tension flange. The center deflection shown in Figure 10 also leaves the straight line at this load which also shows that the girder becomes less effective at this point.

The theoretical compression and tension curves in Figure 10 were computed on the basis that the weld metal and the material adjacent to it were at an initial



COVER PLATE BUCKLING
G-18

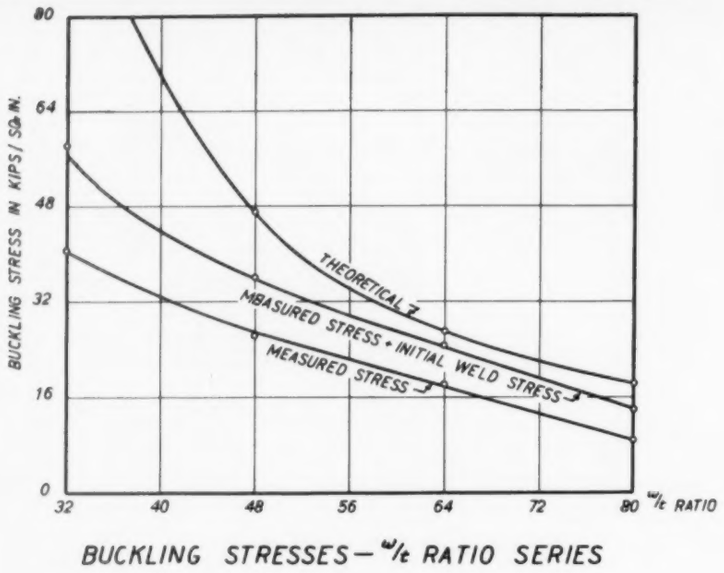
Figure 11

TABLE III

w/t Ratio Tests

1 Girder	2 Maximum load, kips	3 Maximum applied moment, kip-in.	4 Vertical moment of inertia, in.⁴	5 Vertical section modulus, in.³	6 Computed maximum stress, M/S, kips per sq. in.	7 Yield point material, kips per sq. in.	8 w/t	9 Theoretical buckling stress, free edge, kips per sq. in.	10 Stress at which flange plate buckled from stresses, kips per sq. in.	11 Stress at which flange plate buckled from lateral measurements, kips sq. in.	12 Compressive welding stress in flange, kips per sq. in.	13 Internal stress + average measured buckling stress, kips sq. in.
G- 5	32.1	1160	156.4	25.2	46.0	41.4	32	106.0	..	..	17.7	..
G- 6	35.9	1290	197.4	31.9	40.5	40.9	48	47.0	27.0	25.0	10.0	36.0
G- 7	35.9	1290	241.5	39.0	33.1	38.0	64	27.4	18.0	18.0	6.4	24.4
G- 8	37.2	1340	294.8	47.5	28.2	39.4	80	16.9	9.0	8.0	4.8	13.3
G- 9	60.4	2170	347.7	44.9 60.5 55.4	−48.3 −36.0 +39.2	41.5	96	11.7 or 47.0	32.0	32.0	12.0	44.0
G-18	51.3	1850	238.7	38.5	48.0	61.4	64	27.4	22.0	24.0	6.4	29.4
G- 5a	24.7	1330	156.4	25.2	52.7	41.4	32	..	20.0	..	..	..
G- 6a	32.5	1360	197.4	31.9	42.7	40.9	48	..	30.0	..	..	..
G- 7a	33.5	1410	241.5	39.0	36.2	38.0	64	..	24.0	..	..	..
G- 8a*	23.3	980	294.8	47.5	20.4	39.4	80	..	8.0	..	..	..
G-18a	48.9	2060	238.7	38.5	53.3	61.4	64	..	..	..	..	..

*Girder buckled at point where it had buckled in first test.



BUCKLING STRESSES — w/t RATIO SERIES

Figure 12

yield-point stress of 60,000 lb. This results in the net section effect previously mentioned. The resultant compressive welding stress in the flanges necessary to balance the initial tension stresses are given in column 12. Column 13 gives the sum of the initial welding stress plus the measured buckling stress. This stress approximates the theoretical buckling stress of column 9.

The second criterion for determining the buckling deflections was derived from measured local buckling deflections similar to those shown in Figure 11. These values also check the values found in the first method. The load at which the flange plate had buckled a distance equal to one-sixth the plate thickness, was taken as the buckling load. When the local buckling exceeds this value, the compression force passes outside the kern point of the plate, the maximum compressive stress in the plate is then more than twice the average and tension is also present on the other side of the plate. The compressive stress early reaches its yield point at these points of maximum buckling, local plastic yielding occurs, and there is a redistribution of stress. It is probably due to this fact that the Southwell method could not be used to determine the buckling loads in this series, with the exception of G-18, whose yield point was enough higher than the other girders' so that enough buckling readings could be taken below the yield point to determine the buckling stress. This value was 32,000 lb. per sq. in., which compares with the values in columns 9 and 13 for this girder.

The compression flange plates in a box girder are subjected to a more severe buckling condition than the plates in a column. This is due to the fact that as the girder deflects under vertical load, the cover plates are curved and thus the compression stress is eccentrically applied. Local deflection results and a true buckling stress can not be determined. Figure 11 shows the lateral buckling at three loads and the small insert shows the points where the deflection readings were taken. Lines A and G are diaphragm locations. The flange plate buckled in a double wave longitudinally along the girder as shown by the solid curves for line 3 between the diaphragms. Between the webs, the girder deflects in a single wave, as shown by the dotted curves which give the deflections for line C.

The flanges of the test girders buckled at lower loads than predicted by theory, and it is thought that this is due to internal welding stresses. Girder G-5 showed no sign of cover-plate buckling but yielded first as a unit with no local buckling.

In Figure 12, is plotted a curve showing how the measured buckling stresses varied with the w/t ratio. The curve of measured stresses plus welding stresses is also given. This second curve is fairly close to the theoretical.

Girder G-9 as shown by Table III, failed at a higher stress than the other girders, even though it had a higher w/t ratio. The reason for the added strength was the longitudinal stiffener on the compression flange of this girder. The stiffener should have increased the strength of the plate to one having a w/t ratio of 48, and this it did. This girder failed when the stiffener reached its yield point, thus rendering the stiffener ineffective. The three values for the section modulus given in column 5 for this girder are the values for the edge of the stiffener, the compression flange, and the tension flange.

The primary object of the retests in the w/t series

Figure 13

Figure 14

was to determine if loading the girder through the rail would weaken the girder. As is shown in Table III, every girder with the exception of G-8a failed at a greater bending moment than in the first test. Girder G-8a could not be fully straightened from the first test and this probably accounts for its low strength value in the retests.

The measured buckling stresses of the girders in the retests were greater than in the original tests except for G-5a. Local yielding of the rail under the load occurred in this girder and probably lowered the buckling value. With the exception of G-5a, the rail was just laid on the girder and was not fastened to it. It acted as a stiffener by preventing the cover plate from buckling upwards. Since the buckles could only occur downwards, the effect was the same as if the edges of the plate were fixed perpendicular to the cross section of the beam. Loading the girder through the rail caused local bending stress in the flange plate which will be discussed later.

Figure 13 is a picture showing how girder G-6a failed in this series. This picture shows that both the flange and web buckled at the ultimate load. The cover plate necessarily had to buckle inwards with the edges of the buckle perpendicular to the length acting as fixed edges. This picture also shows the details of the loading in the retests.

Figure 14 on the other hand shows the details of the failure when the girders were not loaded through the rail. The difference in the buckles in this picture and in the preceding picture are evident, since in Figure 14 the buckles are alternately up and down.

Figure 15 is a picture showing some of the details of the test set-up for girder G-5.

h/t Ratio Tests—The results of the third series of tests, the *h/t* series, are given in Table IV. The second column gives the maximum loads taken by the girders and the third column gives the maximum moments. Column 6 gives the maximum computed stress as computed by the usual beam formula. With the exception of G-10, these stresses are less than the yield point. However, the actual measured stresses in these girders were at the yield point just prior to failure. The beam formula does not give true values at the higher loads since at these loads the webs had buckled and transferred some of their load to the flanges.

Column 9 gives the theoretical buckling load for the girders with the computations based on free edges. Column 10 gives the measured buckling stress as determined from Southwell's method. This stress is the stress at which the webs would have buckled had they

TABLE IV

h/t Ratio Tests

1 Girder	2 Maximum load, kips	3 Maximum applied moment, kip-in.	4 Vertical moment of inertia, in.^{4}	5 Vertical section modulus, in.^{3}	6 Computed maximum stress, M/S, kips per sq. in.	7 Average yield point, kips per sq. in.	8 h/t	9 Theoretical buckling stress free edge, kips per sq. in.	10 Measured buckling stress, Southwell, kips per sq. in.	11 Measured buckling stress from lateral deflection, kips per sq. in.	12 Measured buckling stress from stress curves	13 Average of columns 11 and 12, kips per sq. in.
G-10	169.0	6090	2953	146.0	41.7	37.3	320	33.2	43.0	..	..	..
G-10a	118.0	4250	2953	146.0	29.1	37.3	320	33.2 6.4*	..	..	..	..
G-11	65.4	2350	876	71.5	32.9	39.4	192	17.6	27.6	19.0	18.0	18.5
G-12	63.3	2280	717	64.3	35.4	37.4	176	21.0	31.0	20.0	20.0	20.0
G-13	56.4	2030	577	56.3	36.0	37.8	160	25.0	34.0	25.0	22.0	23.5
G-14	50.7	1820	456	49.3	36.9	38.7	144	31.3	39.0	29.2	26.0	27.6

*This value computed assuming no stiffener.

been perfectly straight and the material perfectly elastic.

However, the webs were not perfectly straight, there was some eccentricity, so that the webs buckled at lower loads than that given by Southwell's method. Columns 11 and 12 give the measured buckling stresses determined by two methods and column 13 gives the average of column 11 and column 12.

The buckling stress given by Southwell's method is greater than the theoretical buckling stress based on free edges given in column 10. This shows that the edges were not free and that the true buckling stress is

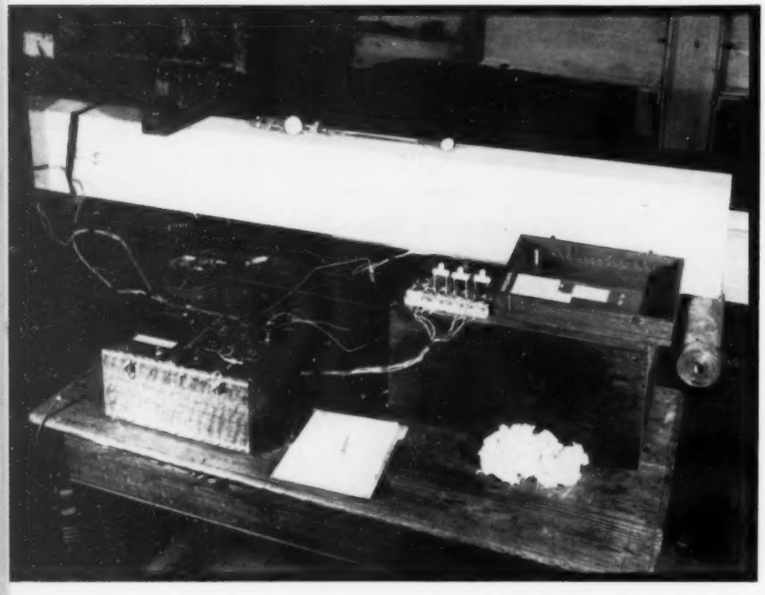


Figure 15

greater than that computed on the basis of free edges. This is shown in Figure 16, which shows the buckling in a vertical plane. The curves on the left side are for the north web and those on the right side are for the south web. The fact that the curves have a point of contraflexure at the bottom shows that there was partial fixity along the edge of the plate.

Figure 17 gives curves for the web buckling along the length of the girder, in the middle third of the girder. This section which had bending stress only and no shear, buckled into five waves in a 6 ft. length. This shows why vertical stiffeners are of little value in preventing buckling due to bending, since they can have little effect unless their spacing is less than the wave length of the buckles. Otherwise the plate would buckle between the stiffeners. Such a spacing of vertical stiffeners is uneconomical.

All girders with the exception of G-10 buckled in a manner similar to that shown in Figure 17. G-10 deflected into a single wave in the 6 ft. length of the center panel, showing that the effect of the horizontal stiffener was to increase the wave length of the buckle. This is the usual effect of longitudinal stiffeners. Girder G-10 was not loaded to failure and the maximum load given for this girder in Table IV is the maximum load which was applied to the girder.

In girder G-10a, where most of the longitudinal stiffener had been burnt off, the web buckled into three waves in the middle panel. Figure 18 gives a picture of this girder after failure.

The buckling criteria applied to the other girders were not used on G-10 or G-10a, since it was thought that the presence of the longitudinal stiffener made the criteria inapplicable. The buckling stress of G-10a was the stress of 29,000 lb. per sq. in., which was present

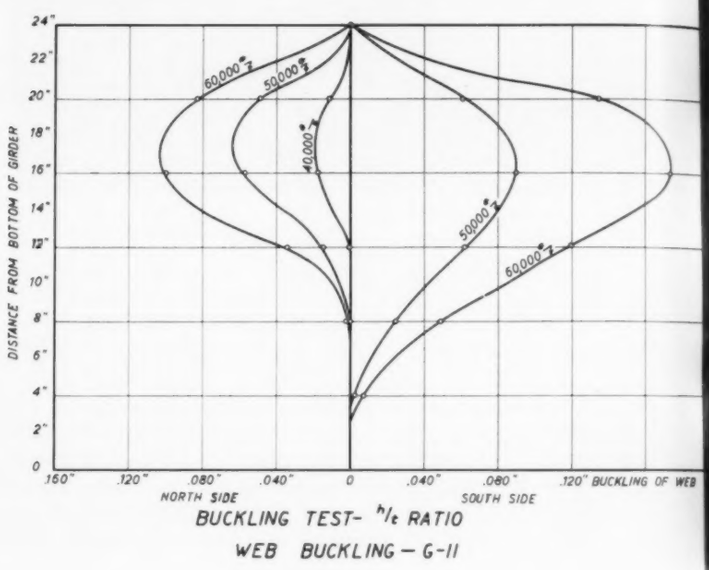
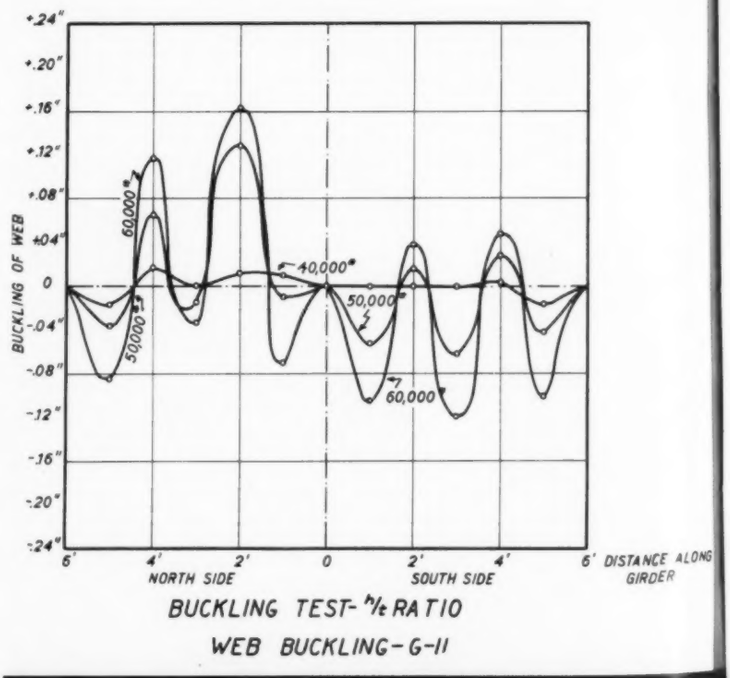


Figure 16

▲

Figure 17



when the girder buckled. This girder failed very suddenly, whereas all the other girders failed gradually. In girder G-10, which was not taken to its buckling load, the stress criteria never indicated buckling, while the buckling stress, determined from a lateral deflection of 0.02 in., was only about 2000 lb. per sq. in. This lateral deflection, which is the distance to the kern of the $\frac{1}{8}$ in. plate, does not apply to the stiffener-plate combination.

It is believed that the stress criteria for buckling is better than the lateral deflection criteria. In plates with very large height-thickness ratios, the load could pass outside the kern point so that tension would be

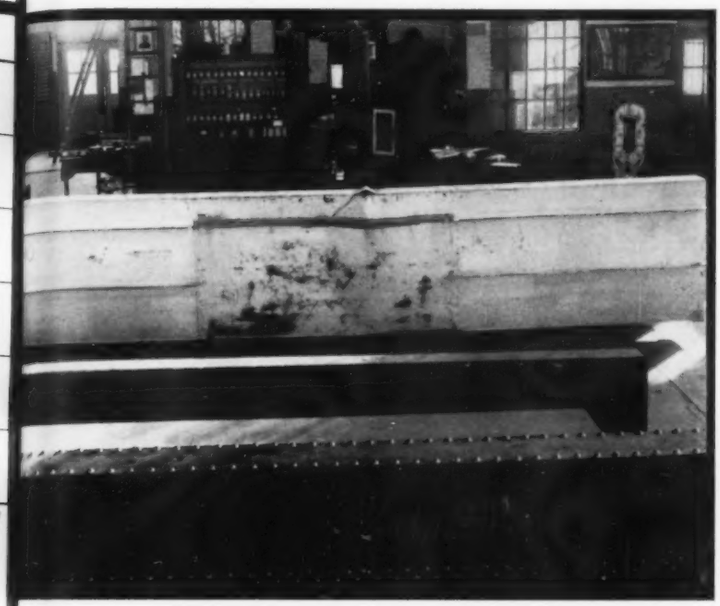


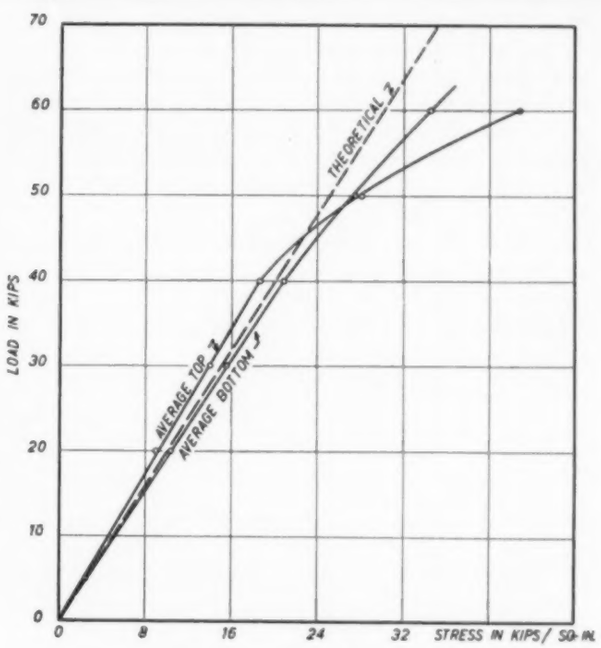
Figure 18

present, and the compression stress would still remain within the elastic limit. However, the proportions of the plates were such with the exception of G-10, that when the deflection had exceeded the kern distance, the maximum stress had exceeded the yield point.

Figure 19 shows the stress curves for G-11 and one can easily determine from the compression curve where the web started to become ineffective. The deflection curves for this girder also broke away from the straight line at this load.

In Figure 20 are plotted the buckling stresses for the various h/t ratios of the web plates. In the right side of the figure the buckling stresses are plotted against the h/t ratio and the result is a curve. In the left side of the figure, the stresses are plotted against $(t/h)^2$ and the resulting curves are straight lines. The left figure is more useful in extrapolating for results outside the test range. Three curves are plotted in this figure, the measured buckling stress by Southwell's method, the theoretical buckling stress assuming free edges and the actual measured buckling stress. The weakening effect of small waves in the plate is quite evident when comparing the measured buckling stress with the stress by Southwell's method. It is also quite probable that there may have been initial compressive stresses in the plate due to the welding which would have contributed to the low buckling values.

The theoretical deflections, including the shear deflection, were computed for the girders in this series and compared with the measured values of the straight line portion of the deflection curve. The ratio between the measured deflections and the theoretical varied from 1.26 in the case of G-10 to 0.99 in the case of G-14. Apparently the lack of straightness or initial waves in



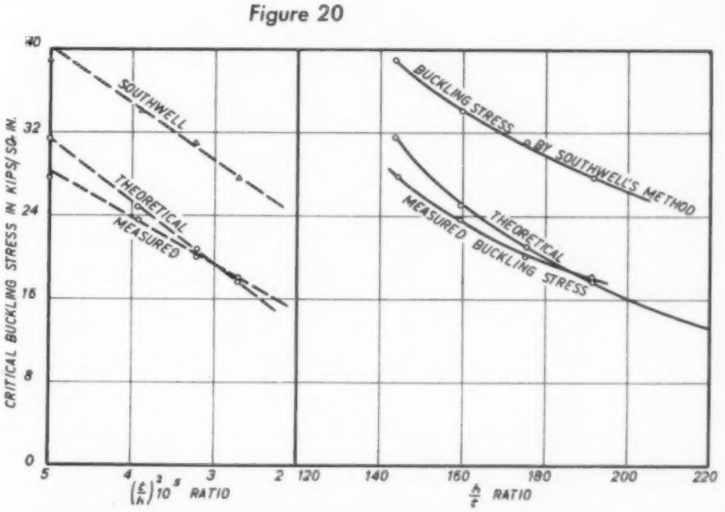
BUCKLING TEST– $^{h}/_{t}$ RATIO

STRESSES G-11

Figure 19

▲

Figure 20



BUCKLING STRESSES– $^{h}/_{t}$ RATIO SERIES

the thin web plates decreased the effective modulus of elasticity and thus increased the deflection.

Miscellaneous Tests—The stresses on a number of diaphragms were measured by means of strain rosettes in which the strains were read on the electric gauge. For the girders in the *l/b* series, no stresses were measured on the diaphragms in the bending tests except when the load was placed directly over the diaphragm. When this was done, the maximum stress measured was 1000 lb. per sq. in. in one corner of the diaphragm for a 3000 lb. load. The area of the diaphragm was 0.75 sq. in.

For G-7, the stresses on a vertical line on one side of the upper corner of an unloaded diaphragm increased by 110 lb. per sq. in. for every 5000 lb. load increment up to 20,000 lb. At this load, the cover plate buckled and then the stress in the diaphragm increased by +1300 lb. per sq. in. for every 5000 lb. load increment up to the ultimate load. The buckling of the cover plate applied a moment to the diaphragm and the bending force set up in the diaphragm reduced the buckling of the cover plate.

Strain rosette readings were also taken on the inside of the cover plate in girders G-5 and G-18. Table V summarizes the results of these tests.

In test No. 2 on G-5, the girder was loaded at the span center by a concentrated load through a 2 in. square steel block on the center of the cover plate. Such a load introduces bending stresses both in the longitudinal direction of the girder and in the cross direction between the webs. The cross stresses given in column 3 are the greater since the span is shorter in this direction. Column 4 gives the longitudinal bending stress. This longitudinal bending stress in the cover plate is additional to the direct compressive stresses in the flange caused by the bending of the whole girder. The stress given in column 4 consists only of the bending stress. It can be seen from columns 5 and 6 that the proportions of the cross bending stress to the longitudinal bending stress is inversely proportional to the ratio of the distance between the webs and the diaphragm spacing when the load is applied half-way between the diaphragms. In column 7 the computed cross stress is given for a concentrated load applied to the plate. This stress is computed from a semi-empirical formula:

$$f = \frac{M}{4SL}$$

where f is the average cross bending stress in pounds per square inch.

M is the fixed end cross bending moment in inch-pounds.

TABLE V

Plate Stresses

1 Girder	2 Test	3 Cross stress kips per sq. in.	4 Longitudinal stress, kips sq. in.	5 Ratio Col. 5 Col. 6	6 Ratio cross span to longitudinal span	7 Computed cross stress, kips per sq. in.	8 Applied load, lb.	9 Per cent of load taken by plate	10 Remarks
									Concentrated load placed:
G- 5*	1	+ 6.0	+2.0	.33	.25	+ 8.0	500	80	At center-line on 1 in. square longitudinal bar.
G- 5	2	+ 5.6	+1.3	.23	.25	+ 5.7	355	98	At center-line of girder on 2 in. block on center of cover plate.
G- 5	3	+ 2.4	+0.5	.21	.25	+ 3.4	640	70	6 in. from diaphragm on 2 in. sq. block.
G- 5	4	+ 0.7	0	...	.25	+ 2.0	370	35	6 in. from diaphragm on 1 in. longitudinal bar.
G- 5	5	+ 5.1	+1.2	.24	.25	+ 63.3	4020	8	On 3 in. 5.7 lb. I-beam at center-line of girder.
G- 5	6	+ 6.6	+0.1	...	.25	+126.0	8000	5	On 3 in. 5.7 lb. I-beam 6 in. from diaphragm.
G- 5	7	+22.0	+4.8	.25	.25	+ 63.3	4020	35	On 2⅜ in. I-beam at center-line of girder.
G-18†	1	+ 8.7	..	...	1.00	+ 7.9	490	110	On 2 in. square block directly over gauge.
G-18	2	+ 5.7	..	...	1.00	+ 4.3	800	130	On 2 in. square block at center-line of girder.
G-18	3	+ 6.3	..	...	1.00	+ 4.3	800	146	At center-line of girder on 1 in. square longitudinal bar.
G-18	4	+ 3.0	..	...	1.00	+ 9.5	590	32	On 1 in. square bar directly over gauge.

*Location of stress reading—Center of cover plate at center-line of girder.
†Location of stress reading—Center of cover plate 6 in. from center-line.

S is the section modulus of the plate per inch of plate.

L is the clear span between the webs.

This formula has been derived on the basis of a parabolic distribution of bending moment ("Structural Behavior of Battledeck Floor Systems," I. Lyse and I. E. Madsen, *Trans. A.S.C.E.*, 1939, p. 264). The maximum stress directly under the concentrated load is three times the average value given above.

When the load is applied to the center through a light rail as in test 1 on G-5, some of the load was transferred to the diaphragms and some of the load was spread a longer distance over the plate resulting in a lower maximum stress. When the load was applied to a heavier rail as in test 7, only 35 per cent of the load was taken by the plate, and when the load was taken by a still heavier rail as in test 5, only 8 per cent of the load was taken by the plate. The other tests show that the maximum stress occurs at the load point, when the load is placed half-way between the diaphragms.

These tests show that the bending stress in the plate can be made quite small by using a stiff rail and by spacing the diaphragms closely together. In order to determine the proportion of the load taken by the plate and rail, the deflections of both must be computed and the loads taken by each will be in proportion to the loads which will cause equal deflection in the rail and plate.

Column 9 gives a comparison between the resisting force or stress in the plate at the point where the stresses are measured, and the stresses which would be present there if the load had been applied to this point as a concentrated load. The value of 146 per cent for test 3 of G-18 merely shows that the bar distributed the load so that the stresses at the gauge point which is 6 in. from the load point, are higher than they would have been if the bar had distributed no stress. In other words, the rail distributes the load.

DISCUSSION OF RESULTS

l/b Ratio Tests—The l/b ratio tests show conclusively that within usual limits of crane design, there is no reason to use a compressive design stress lower than the tension design stress, if the girder is proportioned so that local buckling does not occur in the flange plate. In addition, any local buckling which may occur, does not depend on the span of the crane, so there is no reason in this case, to make a larger reduction in design stress for a long span crane than for a short span crane.

The box portion of the girders tested had a width-depth ratio of one-fourth. This is in line with common practice in crane construction for riveted girders, and is probably smaller than normal proportions for welded girders. The thickness ratios for the webs and flanges in actual cranes may be about three times those in the model girders. However, girders with such ratios would fail due to local web or flange plate buckling rather than buckling of the girder as a whole.

In addition, the torsional constant and the moment of inertia of the girder increase with the square of the dimensions of the girder. Thus when these factors are all considered, full size crane girders will be relatively stronger than the model girders as far as lateral buckling is concerned. In cranes where there is a good end connection between the girder and the end tie, some lateral end fixity is present, and this further reduces the possibility of lateral buckling of the girder as a whole.

Lateral load applied to the girders will lower the buckling load, but for girders with the proportions tested, this effect is not appreciable. The fact that the lateral load did not weaken the girders is borne out by the retests of the l/b series.

The test results show that a maximum allowable l/b ratio of at least 110 for welded girders and 80 for riveted girders could be used without reducing the allowable compressive stresses. Since these ratios are far greater than any used in present practice, it may be advisable to lower this limit to 60 for the allowable l/b. A higher ratio than this will probably be uneconomical, since such proportions result in a low lateral section modulus and consequent high lateral stresses.

In designing boxes of unusual proportions where there may be some doubt as to the lateral strength of the girder, it would be advisable to check the buckling load by the stability formula for the concentrated load at the center of the beam.

$$P_{crit} = \frac{16.93\sqrt{B_1 C}}{l^2}$$

In using this formula, the computed torsion constants

Figure 21

of riveted girders must be reduced by a factor of one-third.

The l/b ratio tests are probably the most important in this whole girder program, since they show where design stresses for box girders may be safely increased, provided proper precautions against local buckling are observed. Such an increase should result in appreciable saving.

Figure 21 is a picture which shows the details of the failure of girder G-1, which girder had the longest l/b ratio. The failure is seen to be local buckling of the compression flange with no signs of lateral buckling.

In discussing this series, it should be noted that the stress curves, such as those in Figure 5, are not actual stresses above the yield point, but rather represent strains.

The tests in this series also confirm the fact that the strain distribution in the girders remained linear over the cross section above the yield point.

The shift in the neutral axis due to the weld stresses was an interesting phenomenon. These stresses might be relieved by annealing or by stressing the girders above the yield point. However, the first method is impracticable and the second method would not occur in service. Where buckling is not a factor, these internal stresses have no appreciable harmful effect.

w/t Ratio Tests—In welding thin plates it is necessary to keep the heat at a minimum, for otherwise the plates may buckle without the application of external load. It follows then, that the welding process may leave internal stresses which are not large enough themselves to cause buckling, but which will cause an apparent low buckling stress, when the girder is loaded. Whether the internal weld stresses will contribute to buckling or not will depend on the location and size of the welds. In a box girder, the tension due to the welds in the corners of the box cause compression stresses in all four plates, since the sum of the internal stresses must equal zero for equilibrium. These stresses will lower the buckling values of the compression flange and web plates.

Since the internal stresses can not exceed the yield point, elastic buckling can never occur if the theoretical buckling stress of the plate is equal to twice the yield strength of the plate. This condition was true in girder G-5 where there was no indication of plate buckling. In the other girders buckling could occur.

It is believed that the difference between the theoretical and buckling stresses in the w/t test series was primarily due to internal weld stresses. These stresses were much larger in the test models than they would be in actual crane girders, since the model sections had a larger proportion of weld metal than is usually used in actual construction. This is due to the fact that the girders had minimum practical size welds.

The weld stresses as computed in the tables are based on the area of the weld metal and the relative proportions of the flanges and webs. They were not actually measured in the girders, but the data by which they were computed were determined from the shift in the neutral axis, and from pilot models in which the internal stresses due to welding were measured with a 10 in. Whittemore strain gauge. These computed stresses may be somewhat in error quantitatively but should represent actual conditions qualitatively. In riveted girders, these stresses would not occur.

The criteria used to determine the buckling stresses may not be strictly correct mathematically, but it is felt that they define the very practical point where the plates cease to be fully effective due to excessive lateral deflection.

The buckling stresses, measured in this series, were less than the theoretical values as shown in Figure 12, but in larger girders where internal weld stresses are much smaller, or in riveted girders, the theoretical values for buckling stress should be close to the actual values. It is believed that these theoretical values can be used safely in design. This is particularly true when the load is applied to the girder through a rail, since the rail acts as a stiffener to the girder.

The rails must be large enough to transfer most of the load to the diaphragms, or otherwise bending stresses in the cover plate will weaken the cover plate.

When an adequate stiffener is applied to the flange plate, the w/t ratio of the plate can be taken as the distance between the web and stiffener. This was borne out by the test on G-9.

The alloy steel girder was somewhat stronger than the same carbon steel girder, since local yielding due to small buckles in the plate occurred more rapidly in the carbon steel girder than in the alloy steel girder.

h/t Ratio Tests—One of the standard references on stability is "Theory of Elastic Stability" by S. Timoshenko. Some of this material is in a form which is inconvenient for the use of the designer. A recent paper by L. S. Moisseiff and F. Lienhard ("Theory of Elastic Stability Applied to Structural Design," *Proceedings, A.S.C.E.*, January 1940) is an attempt to correlate stability theory so that the designer may use it conveniently. Most of the following theory comes from this paper. The general formula for buckling of plates is:

$$\sigma_{crit} = K \frac{\pi^2 E \zeta}{12\left[1-\left(\frac{l}{m}\right)^2\right]}\left(\frac{t}{d}\right)^2$$

in which t = the thickness of the plate.

d = the width of the plate.

$\frac{1}{m}$ = Poisson's ratio.

σ_{crit} = the buckling stress.

E = the modulus of elasticity.

K = a constant which depends on the dimension of the plate, the end conditions, and the kind of stress.

ζ = the modulus factor, a value by which E is multiplied when the stress exceeds the elastic range.

For plates with simply-supported edges and subjected to pure compression, the value of K is 4. For plates with simply supported edges, in pure bending, the value of K is 24.

For freely supported plates, subjected to a combination of pure compression and bending, the value of K is given by the empirical equation:

$$K = \alpha^3 + 3\alpha^2 + 4$$

The value of α is given by the equation:

$$\sigma_y = \sigma_o\left(1-\alpha\frac{y}{d}\right)$$

in which y = the distance of stress from the edge.

σ_o = unit stress at $y = 0$, the upper or lower edge of the plate.

σ_y = unit stress at a distance y from edge.

When a crane girder is subjected to both lateral and vertical loads, the web plates are subjected to both a compressive and bending stress and the allowable stress must be computed from the above formulæ. Since the combination of vertical and lateral loads results in a lower buckling stress for the same h/t ratio, this factor should govern in setting the allowable h/t ratio.

The test girders in this series subjected the webs to pure bending only. As is shown by the values of Table IV, and the curves of Figure 20, the measured buckling stresses agree fairly well with the theoretical. The computed theoretical stress was based on simply-supported edges, but no account was taken of the ζ factor which reduces the theoretical buckling stress at stresses above the elastic limit (Table 3, "Theory of Elastic Stability Applied to Structural Design," L. S. Moisseiff and F. Lienhard, *Proceedings, A.S.C.E.*, 1940). If this factor is taken into account, the theoretical stresses check the measured stresses very closely.

The measured buckling stress by Southwell's method showed that there was some partial fixity on the edges of the plate. However, the small waves in the plate and possibly the welding stresses reduced the buckling stress to the simply-supported edge value. For a maximum fibre stress of 18,000 lb. per sq. in. and a safety factor of 1.83, the ratio between 33,000 and 18,000, the maximum allowable h/t ratio is about 144. If the girder is subjected to lateral moments as well as vertical moments, this ratio should be still less.

Apparently h/t ratios used in present cranes are too high if design stresses are to be increased. These ratios have not caused any trouble in the older cranes since the maximum compressive design stresses in these cranes seldom exceed 12,000 lb. per sq. in. Furthermore, the webs of these cranes may sometimes buckle slightly in service with no damage at low design stresses. As was shown in Table IV, there is a large margin between the load at which the web buckled and the load at which the girder failed. For this reason, a relatively smaller factor of safety can be used in web design. If high web-thickness ratios are to be used with high stresses, longitudinal stiffeners should prove economical. The effectiveness of longitudinal stiffeners was clearly demonstrated by girder G-10. The paper by Moisseiff and Lienhard also gives data relative to such design procedure.

Miscellaneous Tests—When the load was applied directly to the flange plate of a box girder, the load was transferred to the webs by cross bending in the flange plate. There is also a longitudinal bending stress in the flange plate which is inversely proportional to the diaphragm spacing and the distance between the webs. This longitudinal stress will reduce the buckling strength of the flange plate appreciably. For this reason, it is essential that crane girders have rails that are strong enough to transfer the wheel load to the adjacent diaphragms and are sufficiently stiff so that the rail will not deflect enough to transfer appreciable bending stresses to the cover plate.

CONCLUSIONS

The box girder tests made in this investigation lead to the following conclusions:

1. Box girders of the proportions tested will fail at stresses as computed by the ordinary beam formula, which are about 20 per cent higher than the yield point of the material, unless local buckling first occurs.

2. For the girders tested, there was no tendency for lateral buckling to occur, even when the girders were loaded with both a vertical load and a 10 per cent lateral load applied to the top flange.

3. For welded girders with l/b ratios up to 110, and riveted girders with l/b ratios up to 80, it is not necessary to reduce the allowable compressive design stress.

4. The strain distribution of the girders was linear over the cross section of the girder up to the ultimate load.

5. There was a shift of the neutral axis towards the compression side of welded girders as the load was applied, apparently caused by welding stresses.

6. Internal weld stresses lowered the buckling strength of the model girders. These stresses may be small in full-size cranes.

7. The buckling stress of the cover plate of a crane box girder can be closely computed from the theoretical buckling formula or the curve in Figure 12, which assumes simply-supported edges for the plate.

8. The rail of a crane acts as a stiffener on the cover plate.

9. The allowable w/t ratio should be computed using the distance between the stiffeners when stiffeners are used. The stiffener must be rigid enough so that it will not buckle.

10. The alloy steel girder was somewhat stronger than the carbon steel girder because it did not yield locally as rapidly as the carbon steel girder.

11. The buckling stress of webs subjected to pure bending can be determined from the theoretical buckling formula using the K for free edges. These values, disregarding ζ are given in Figure 20. If the girder web is subjected to both compression and bending, the values in Figure 20 for the h/t ratio should be reduced.

12. The present allowable h/t ratios must be reduced if design stresses are to be increased.

13. Longitudinal web stiffeners are very effective in preventing buckling of the web. For high h/t ratios, consideration should be given to the use of such stiffeners.

14. Vertical stiffeners are of little use in preventing compression web buckling.

15. A smaller factor of safety may be permissible in computing plate thickness ratios since there is a great deal of reserve strength in the girders after the plates buckle.

16. The stress in the unloaded diaphragms of the girders in the bend tests was negligible. The stresses in the loaded diaphragms were small.

17. The diaphragms reduced the buckling deflection when the flange plates buckled in a girder. When this type of buckling occurred, a small stress was present in the diaphragm.

18. Crane girders must have rails stiff enough to transfer the wheel load to the diaphragms without causing appreciable stresses in the cover plate. Otherwise the strength of the cover plate will be reduced.

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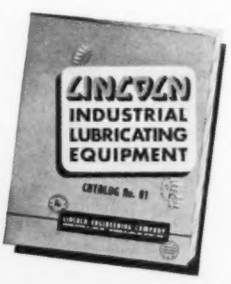

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CARNEGIE-ILLINOIS STARTS NEW ELECTRIC FURNACES

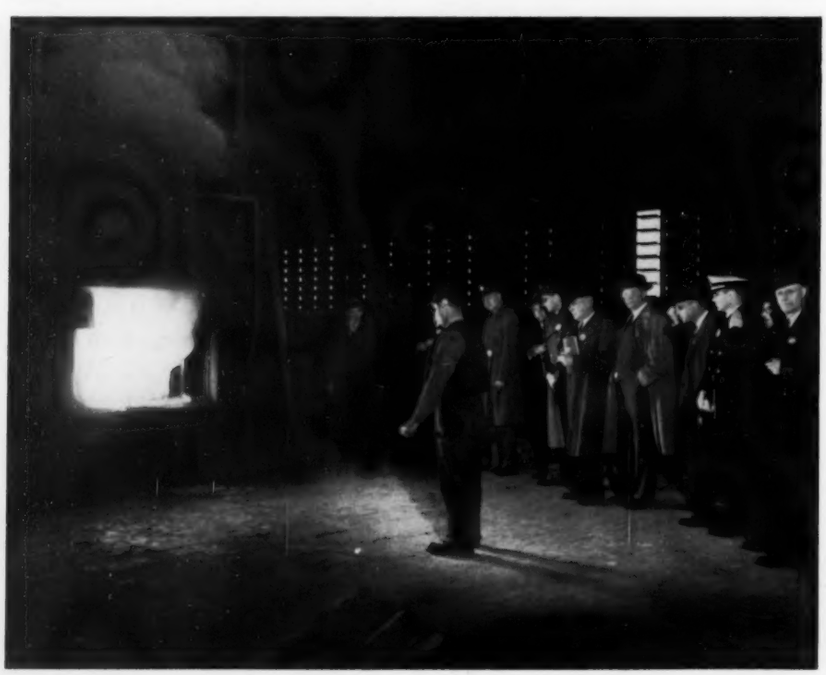
▲ The strain placed upon stainless and alloy steel capacity of the country because of the constantly increasing demands of national defense will be somewhat relieved as a result of the recent start of production of two new electric furnaces at South Chicago works of the Carnegie-Illinois Steel Corporation, United States Steel subsidiary. The plant is the center of United States Steel's stainless steel production, and the new furnaces will increase electric steel capacity by 100,000 net tons annually.

First heats were tapped from the furnaces in the presence of defense and company officials. Among those invited were General J. M. Cummings, Commander, 6th Corps Area, U. S. Army; Homer Hartz, Chicago District Coördinator of the Office of Production Management; Commander J. K. Esler, Great Lakes Naval Training Station; Colonel D. Armstrong, Chicago Ordnance District; and Colonel Reginald Whitaker of the Corps of Engineers.

The cost of the new furnaces, which will more than double the annual South works capacity for the production of stainless and alloy steels, was borne by Carnegie-Illinois. The new furnaces represent the first Chicago district expansion project to be completed by Carnegie-Illinois in its program designed to keep pace with defense needs. Construction of the furnaces was started last March.

The furnaces have a nominal rated capacity of 70 and 30 tons per heat, respectively. These units represent the most recent developments in design, installation, and operation, including facilities for charging the required raw materials. The furnaces are housed in two parallel buildings, one 80 x 850 ft. and the other, 75 x 450 ft. Electric power for the units is supplied from the plant's own generating stations, and delivered to the furnace transformers at 22,000 volts. The larger furnace is equipped with a 15,000 kva. transformer and the smaller with a 12,000 kva.

Defense and company officials watch first helper, Antonio Matuszewski, as he looks into one of the two new electric furnaces which began production October 28, 1941 at South Works of the Carnegie-Illinois Steel Corporation. The officials present are: *(left to right)* J. H. Eisaman, assistant superintendent of electric furnaces, South Works; Col. Barrett Rogers, Chicago Ordnance District; M. F. Yarotsky, superintendent of steel production, South Works; Lt. R. N. Voigt, Chicago Ordnance District; W. E. Hadley, manager of operations, Chicago District, Carnegie-Illinois Steel Corporation; Walther Mathesius, vice president, United States Steel Corporation of Delaware; B. M. Livezey, general superintendent, South Works; Lt. C. D. Kuhn, United States Navy; C. F. Frye, group resources engineer, Office of Production Management; and George Gustafson, assistant superintendent of steel production, South Works.



NEW BLOOMING MILL FOR ALUMINUM PLANT

▲ To supply stock for the manufacture of forgings for airplane motors and fittings, and rod, bar and wire for national defense industries, R. T. Whitzel, superintendent of the Massena works of Aluminum Company of America, has been advised that the company plans to erect a second blooming mill there. Early in Septem-

ber, OPM stated that this expansion was essential to the defense program.

The blooming mill and its necessary facilities will be housed in steel and brick buildings covering an area of more than 450,000 sq. ft. A completely new melting department will be established to serve the blooming mill.

The mill and buildings are now being designed by the company's engineering department in Pittsburgh and will be erected by the company's own construction forces. The grading of the site has already started and the company is exerting every effort to speed delivery of materials and equipment, Mr. Whitzel said.

The blooming mill now being developed for location at Massena as well as the present blooming mill may be considered the mother plant of the Aluminum Company's entire forging operations, according to Mr. Whitzel, since 60 per cent of their output will be used in making forging stock. Airplane rivets are another important defense item whose manufacture begins in Massena in the blooming mills.

Largest of the buildings for the new mill will be approximately 160 ft. by 1400 ft. The buildings will be similar in design and construction to the present blooming mill buildings; while the mill itself will be of the latest type and have incorporated in it the improvements resulting from the company's past experience in blooming mill operations.

Mr. Whitzel estimates that the mill will cost in excess of $15,000,000. This, he said, is over and above the company's $200,000,000 self-financed expansion program for national defense.

United Engineering and Foundry Company, manufacturers of the new blooming mill, will complete the rolling mill in eight months. The equipment consists of a 38 in. two-high reversing mill, complete with entry tables, mill tables and front and back electric manipulators.

Two unusual features of the auxiliary equipment will include a 3,000,000 lb. motor-operated shear and the piling mechanism. The shear will be of the up-and-down-cut style and the largest of its type ever built by United Engineering. Because of the character of the product, the piling mechanism will be much more elaborate than ever attempted heretofore.

. . Pittsburgh to Texas by Barge . .

"Old Man River" will roll this rolling mill machinery from Pittsburgh to New Orleans, from where it will be towed to Houston, Texas. This barge load, amounting to 13 railroad carloads, is part of the mill equipment being furnished by United Engineering and Foundry Company for the new plant of Sheffield Steel Corporation, at Houston.

IMPROVED TAP CHANGER FOR TRANSFORMERS

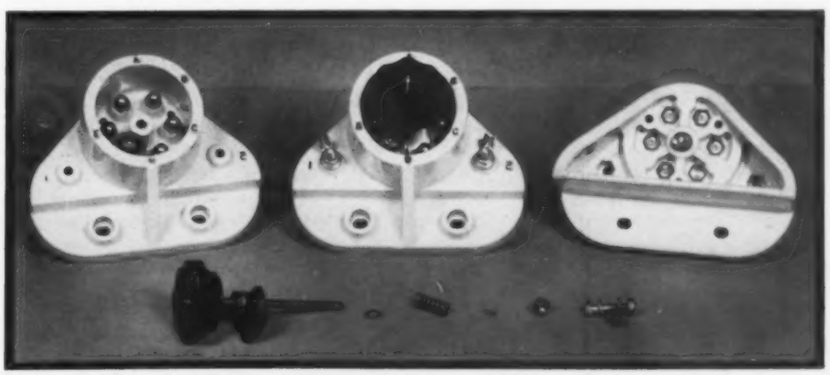
▲ Pennsylvania Transformer Company, Pittsburgh, Pennsylvania, has designed a new improved tap changer which will be available on all distribution transformers. This new tap changer, made of white porcelain, permits greater visibility. All tap positions are marked with black enamel figures on white, thus eliminating any possibility of making the wrong connections.

Engraved on the black bakelite handle knob is a white arrow, which points to the position the tap changer is in. The knob is designed to permit a good hand and finger grip.

The contact points of the tap changer are mounted upon raised bosses. This allows additional creep-

(*Please turn to page 108.*)

In this tap changer, all live parts of leads are recessed into the porcelain body.



TRI-CLAD protection is

REG. U.S. PAT. OFF.

TRI-CLAD Sleeve-bearing Polyphase Industrial Motor Is Most Widely Used

A well-protected, open motor for industry's general-purpose needs. Horizontally mounted—for direct-connected and belt drives—it is economical to operate and suitable for a variety of applications.

TRI-CLAD Ball-bearing Polyphase Motor

Has additional advantage of being mountable in other than horizontal positions. Will take end thrust—for example, from beveled-gear pinion. Similar to the sleeve-bearing motor and, like it, available in many types.

TRI-CLAD Splashproof Ball-bearing Polyphase Motor

For use in wet surroundings, such as dairies, breweries, paper mills, canning factories, etc. Furnished with cast-iron, waterproof conduit box, deflecting end shields, and moisture-resistant insulation.

TRI-CLAD Capacitor-Motor (in sleeve-bearing or ball-bearing types)

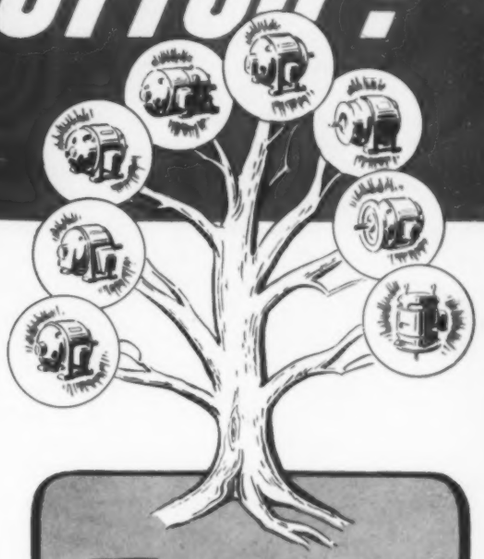
For single-phase operation. Available in types to drive such devices as compressors, pumps, fans, etc. No radio interference; no brushes to wear; quiet operation.

TRI-CLAD Gear-Motor

For economical, compact, low-speed direct or pinion drive. Wide range of output speeds available. Open, splashproof, and capacitor-motor construction. Oil- and dust-tight housings reduce maintenance.

Built for Protection First.....to Last!

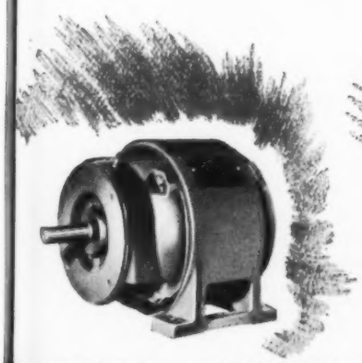
n is now a *family affair!*



THE improvements in Tri-Clad motor design are *fundamental* improvements, not just "sales features" to popularize a new model. That's why they are being extended, month by month, to a whole family of G-E integral-horsepower motors—both general-purpose and special types.

So, in these special fields you can now count on getting the same benefits of extra protection that have made the *general-purpose* Tri-Clad motor the acknowledged leader in performance and service life.

All these new members of the Tri-Clad family are the result of basic redesign to meet modern industrial conditions. Each has new performance and convenience features important to its particular field of service. For complete information on the right Tri-Clad motor for *your* application, consult our local office, or write General Electric Company, Schenectady, N. Y.



TRI-CLAD Ball-bearing Induction Motor, with Face-type End-shield Mounting

For close-coupled attachment to machine tools, compressors, pumps, etc. Motor bolted from driven machine. A few standard mounting dimensions apply to many motor ratings.

TRI-CLAD Round-frame, Ball-bearing Induction Motor, with Flange-type End-shield Mounting

For close-coupled attachment, or direct bolting to driven machine. Mounting dimensions are standard, but larger than those of face-type end shield.

TRI-CLAD Vertical Motor

For general-purpose fan, pump, and machine drives in vertical position. Openings protected and bearings designed for vertical operation. Both polyphase and capacitor-motors available with variety of bases.

Today a wider range of your special needs can be filled by the growing Tri-Clad motor family. Every member offers extra strength and longer life because of these 3 "extras":

Extra Protection

AGAINST PHYSICAL DAMAGE

Sturdy, cast-iron frame and end-shield construction.

Extra Protection

AGAINST ELECTRICAL BREAKDOWN

Stator windings of Formex wire, "armored" by synthetic resins against moisture, heat, oil, and abrasion.

Extra Protection

AGAINST OPERATING WEAR AND TEAR

New sleeve-bearing design and improved ball-bearing mounting lengthen life.

In addition, you'll find the modifications to meet special requirements are soundly engineered to give you space-saving, time-saving, and money-saving advantages all down the line. Consult your G-E representative for Tri-Clad horsepower ratings now available.

GENERAL ELECTRIC

750-90-8058

(*Continued from page 105.*)
age between adjacent stationary contact points.

All live parts of the leads are recessed into the porcelain body of the tap changer so as to obtain greater creepage distance between adjacent leads. For added protection these leads are insulated with varnished cloth tubes.

There are two studs on each side of the tap changer. These serve to make the connection between the high voltage leads from the coil and the line leads coming through the bushings. They are also numbered in black and white.

The heavy porcelain skirt, which is part of the tap changer body, provides additional insulation between the leads and the core. This added protection eliminates any possibility of the leads becoming grounded.

GALVANIZERS' MEETING HELD IN PITTSBURGH

▲ The tenth meeting of the Galvanizers Committee, sponsored by the American Zinc Institute, was held at the William Penn Hotel, Pittsburgh, on Friday, October 31. More than fifty representatives of the member companies attended the meeting.

The chairman of the committee's governing board, N. E. Cook of Wheeling Steel Corporation, opened the meeting at 9:30 A.M. and welcomed the delegates. The round table discussion at the morning session was led by D. A. Russell of Youngstown Sheet and Tube Company; F. F. Aloi of Bethlehem Steel Company acted as chairman. At the afternoon session, presided over by F. G. White of Granite City Steel Company, there was further discussion of topics vital to the interest of galvanizers during the National Emergency.

The governing board held its annual meeting and announced the election of B. P. Finkbone of American Rolling Mill Company, and D. A. Russell to serve as members of the governing board for a three-year term to succeed R. H. Dibble of Carnegie-Illinois Steel Corporation, and F. G. White, whose terms automatically expired. N. E. Cook was re-elected chairman.

An informal dinner at the William Penn concluded the sessions.

While the date of the next meeting was not definitely set, it was decided to hold it in conjunction with the annual meeting of the American Zinc Institute next April.

NEW ALUMINUM MILLS UNDER CONSIDERATION

▲ Dispatches from Washington state that negotiations are under way between the Defense Plant Corporation and United Engineering and Foundry Company for the construction of two aluminum rolling mills on the Pacific Coast. Totaling $40,000,000, the contracts are believed to call for United to act as general contractor and engineer on the mills, in addition to constructing the equipment. Sites for the plants have not been selected, but the general areas of Portland, Oregon, and Los Angeles, California, are under consideration.

FIRM CELEBRATES 25th ANNIVERSARY

▲ The Amsler-Morton Company, Pittsburgh, Pennsylvania, has recently published a large 24-page booklet marking the 25th anniversary of the company. Largely pictorial, the booklet presents scenes from a 25 year

ALL THIS at a single glance
HAYS
1. RATE OF AIR FLOW
2. RATE OF FUEL FLOW
3. RATIO OF AIR TO FUEL OR PERCENTAGE OF AIR EXCESS OR DEFICIENCY

USED on multiple zone slab heating furnaces, for instance, to secure maximum heating rate and efficiency of fuel consumption at the same time making possible the control of the furnace atmosphere to be oxiding (excess of air) or reducing (deficiency of air) as dictated by the requirements.

Many other uses in steel mills, gas producing plants, heat generating plants, CO_2 generating plants, sewage plants, chemical plants, kilns, air conditioning—any place where there exists a relation between any two of the following functions: flow of gas or flow of oil or other measurable fluids; pressure, draft, suction, temperature (up to 1000°F.) speed in r.p.m. or inches per minute, position, level and others.

The cross-pointer feature offers an ingenious method of revealing relationships not apparent in separate instruments or conventional scales. This distinct innovation is found only in Hays Visio Ratio Gage.

If you will write us, stating the condition, we will tell you exactly how the Hays Visio Ratio Gage can be applied to your problem.

Address us . . . . 960 Eighth Avenue, Michigan City, Ind.

The HAYS CORPORATION
SINCE 1901 COMBUSTION INSTRUMENTS AND CONTROL
MICHIGAN CITY, INDIANA, U.S.A.

Large Casting for Coast Defense Gun

Major G. H. Knode, chief of artillery inspection, Pittsburgh ordnance district, and J. R. Patterson, vice president of Mackintosh-Hemphill Company, are shown inspecting a side frame section for a 16 in. coast defense gun at Mackintosh-Hemphill's Midland, Pa. plant.

Scenes at this big Midland plant, birthplace of America's largest guns, are a good indication of why steel has become so precious. All the mounting sections are cast here, 10 pieces per gun, including side frames like the one shown above. Together they weigh 420,000 pounds. When the gun is mounted on them, the total weight is 702,000 pounds, enough steel to make approximately 350 autos, or 31,000 refrigerators or 4600 washing machines.

story of engineering progress. Photographs and general arrangement drawings clearly show Amsler-Morton furnaces and equipment for the steel industry, the glass industry and miscellaneous applications. A history of the company and photographs of the personnel are also included.

PROGRAM AT GARY TO INCREASE CAPACITY

▲ A $15,000,000 expansion program to provide increased steel and iron making capacity at the Gary, Indiana, Steel Works for national defense requirements was announced by Carnegie-Illinois Steel Corporation, a subsidiary of the United States Steel Corporation. While this program constitutes a part of the substantial expansion of the country's steel producing facilities recently recommended by the Office of Production Management, the cost of the new facilities at Gary will be borne by the Carnegie-Illinois Steel Corporation.

Work will begin immediately on construction of a new open hearth furnace, the rebuilding and enlargement of a blast furnace, installation of greater soaking pit capacity, the rebuilding of a battery of coke ovens and provision of additional ore unloading facilities. The new facilities are to be completed as soon as possible.

The new open hearth furnace, to be installed in the No. 5 open hearth department at Gary, will provide an increased annual ingot production of 300,000 net tons. Enlargement of Gary Works blast furnace No. 7 to an annual capacity of approximately half a million tons will double its present capacity.

Seventeen rows of soaking pit furnaces will be replaced to increase their capacity from two million to two and a half million net tons annually. The 70 ovens of by-product coke oven battery No. 6 will be rebuilt to provide an annual capacity of 360,000 net tons of coke. The ore unloading capacity of the plant will be increased from 3000 to 4500 gross tons per hour.

NEW PLANT FOR AMMONIUM SULPHATE

▲ The Donner Hanna Coke Corporation has awarded a contract to the Otto Construction Corporation, of New York, for a complete new unit for the production of ammonium sulphate. Construction work will be undertaken immediately.

ANNOUNCES BLOWER TYPE ELECTRIC HEATER

▲ The Edwin L. Wiegand Company announces a new, improved blower type heater which is rated 1500 to 4000 watts, for operation on 115 or 230 volts, 60 cycle, a-c., and 115 or 230 volts, d-c. Chromalox koilstrip heating elements are used in conjunction with quiet cadmium plated fan, driven by enclosed type electric motor, providing an air temperature rise of 32 to 75 degrees F., at an air velocity of 130 to 180 ft. per min., depending upon rating. Designed for portable

This new blower type heater, rated 1500-4000 watts is designed for portable or permanent mounting.

or permanent wall mounting. Adjustable louvres, gray crinkle lacquer finish. Manual control switch (except d-c.) and positive acting thermostatic switch in heater circuit for protection against over-heating if normal temperatures are exceeded.

GUIDELESS MACHINE FOR STRAIGHTENING BARS AND TUBES

▲ A new guideless seven-roll straightening machine for bars and tubes was recently developed by the Sutton Engineering Company for use in the ferrous and non-ferrous industries.

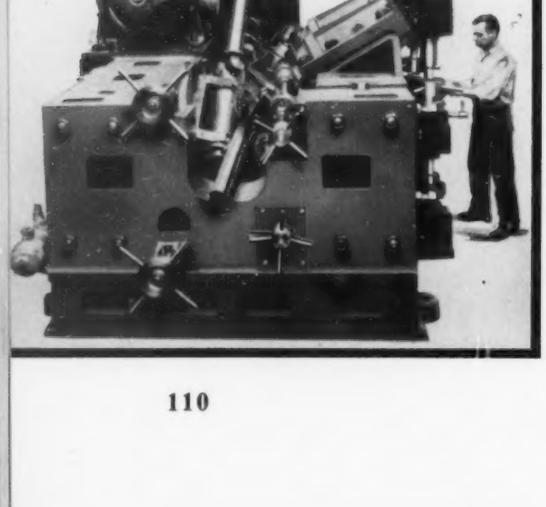
The main purpose in mind in developing this machine was to eliminate the necessity of guides, which is a major problem to all users of rotary bar and tube straighteners. Elimination of guides is accomplished by grouping three rolls, one driven and two idlers, in such a way that the bar or tube is supported at three points. The machine consists of two groups of three rolls with an intermediate pressure roll. As in the standard five-roll machine, straightening occurs in the two sets of cross rolls as well as by using the middle idler roll. In addition to the elimination of guides, this grouping of rolls has many other advantages.

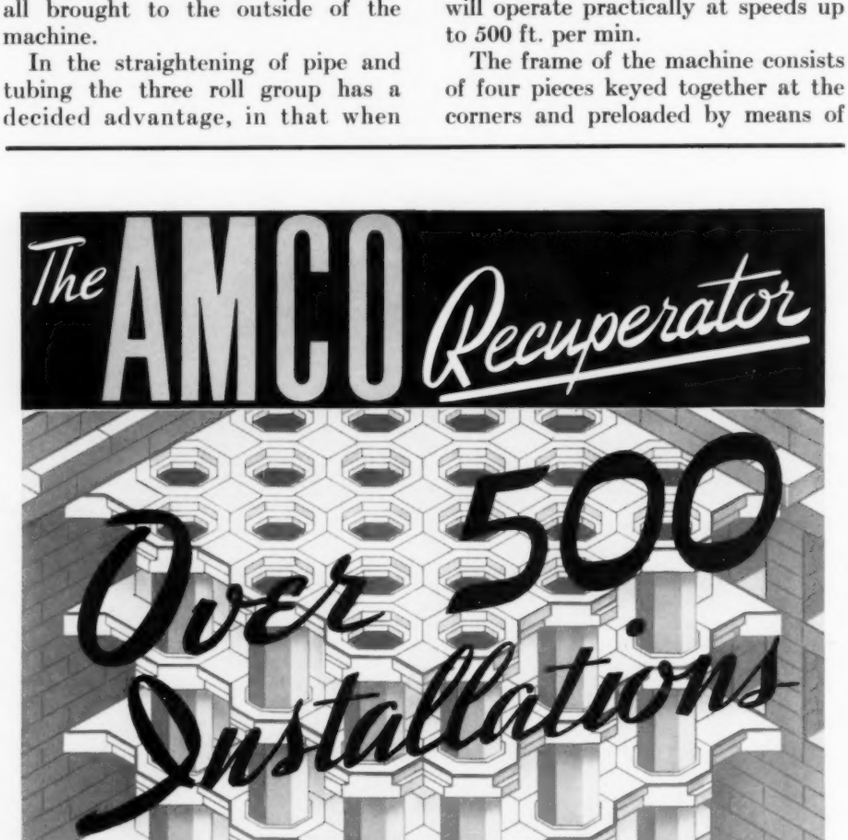
On this machine a wide range of sizes can be accurately straightened from end to end without altering the angle of any of the seven rolls, for example, sizes from 2 in. to 3½ in. can be straightened on the same setting. On larger sizes a greater range is possible. Of course, this greatly reduces the set-up time. However, when it is necessary to alter the angle of the rolls it is a comparatively simple operation, as the adjusting wheels are all brought to the outside of the machine.

In the straightening of pipe and tubing the three roll group has a decided advantage, in that when straightening pressure is applied the amount of ovaling of the tube or pipe is at a minimum since the material is supported at three points. In the polishing of turned and drawn production this machine is very effective, since the material contacts highly finished roll surfaces only. This machine will operate practically at speeds up to 500 ft. per min.

The frame of the machine consists of four pieces keyed together at the corners and preloaded by means of

Guides are eliminated from this straightener by grouping three rolls so that the bar or tube is supported at three points.





large set screws. The two large rolls are driven by means of universal couplings of the most modern type equipped with roller bearings. The reduction drive is totally enclosed, running in oil equipped complete with roller bearings, all of which are lubricated by a splash system which means that all driving gears and bearings are a self-contained unit requiring no manual lubrication. Gearing consists of two high speed bevel gears at motor shaft, a series of single helical reduction gears, and two wide face spur pinions on the roll shafts, meshing with two spur gears. The wide face pinions allow for the slight movement due to the change of angle of the two driven rolls. This adjustment is very slight as the drive shafts are in line at the mean angle of the rolls which is 25 degrees. All gears are forged steel with cut teeth. All seven rolls are equipped with the most suitable type of roller bearings. Bearings other than in the reduction drive are equipped with alemite fittings for grease lubrication and if desired, machine can be furnished with a one shot lubricating system. All rolls are easily removable and may be taken out of the machine without disassembling frame.

PROTECTIVE SYSTEM FOR ROLL NECK BEARINGS

▲ The Brown Instrument Company, Philadelphia, Pennsylvania, announces the development of the "Brown Protection System for Roll Neck Bearings." Present demands being made upon the steel industry for more and more tonnage is resulting in higher rolling speeds which, in turn, means higher bearing temperatures. Should safe bearing temperatures be exceeded only for a short period, burnouts occur, resulting in expensive shutdowns. This can be prevented by the Brown protection system for roll neck bearings.

The complete system consists of three essential parts: (1) the temperature detection elements, (2) the temperature recording system and (3) the signaling or alarm system. The detection elements are rugged pencil type iron constantan thermocouples having an outside diameter of .405 in. The thermocouples are held in position by a specially designed "quick disconnect" fitting which permits instant removal, merely by a slight pull and quarter turn.

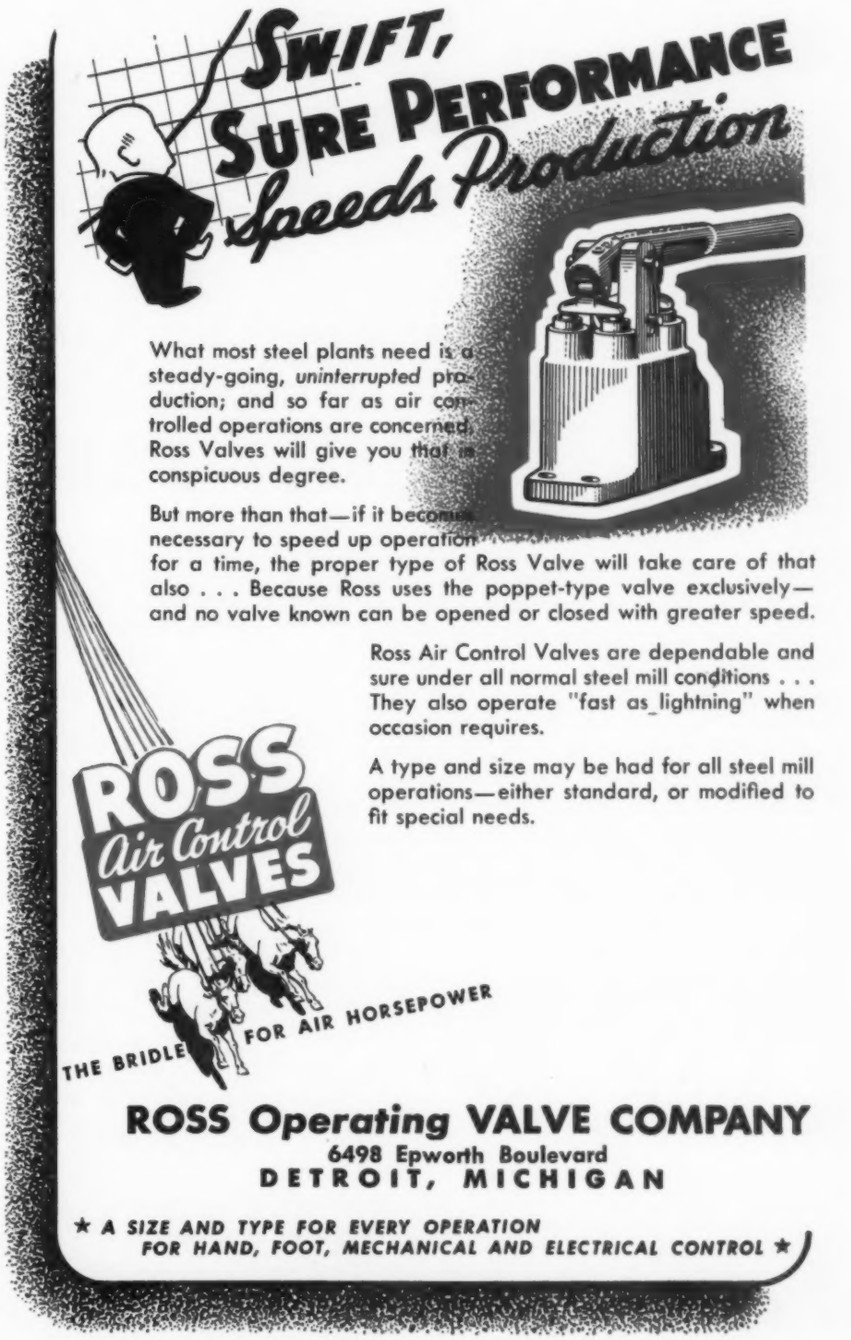
The temperature recording system consists of a Brown multiple record potentiometer which prints each bearing temperature in distinctive colors on a 12 in. wide chart. Balance step rather than fixed cycle printing is used so that each temperature is recorded rapidly. The signal system consists of a relay, signal light, howler or other signal device and a push button for resetting the relay.

Following are a few of the many advantages which accrue from the use of this system:

(1) Provides a continuous, positive, and foolproof indication that the temperature of each bearing in the system is within its safe operating limit.

(2) Warns the operators, by means of a howler or other signal, when any bearing exceeds its safe operating limit.

(3) Indicates, by means of a signal


SWIFT, SURE PERFORMANCE Speeds Production
What most steel plants need is a steady-going, uninterrupted production; and so far as air controlled operations are concerned Ross Valves will give you that in conspicuous degree.
But more than that—if it becomes necessary to speed up operation for a time, the proper type of Ross Valve will take care of that also . . . Because Ross uses the poppet-type valve exclusively—and no valve known can be opened or closed with greater speed.
Ross Air Control Valves are dependable and sure under all normal steel mill conditions . . . They also operate "fast as lightning" when occasion requires.
A type and size may be had for all steel mill operations—either standard, or modified to fit special needs.
ROSS Air Control VALVES
THE BRIDLE FOR AIR HORSEPOWER
ROSS Operating VALVE COMPANY
6498 Epworth Boulevard
DETROIT, MICHIGAN
★ A SIZE AND TYPE FOR EVERY OPERATION FOR HAND, FOOT, MECHANICAL AND ELECTRICAL CONTROL ★

light, on which roll stand the excessive temperature exists.

(4) Shows, by means of a chart record, which bearing in the stand is too hot.

(5) Permits the use of maximum reduction per stand by keeping the operators informed at all times of bearing temperatures, thus speeding up production.

(6) Eliminates unnecessary roll changes due to high bearing temperatures because the operators are warned of the approach of such a condition before actual damage occurs.

(7) Reduces the maintenance cost of roll necks because less re-machining is necessary.

(8) Lengthens the useful operating life of the bearings.

Even in Winter
Hot Spots Slow Up
Production . .

TRUFLO CRANE CAB FANS
For cooling interiors of crane cabs and other confined areas. Adjustable both horizontally and vertically. Four blade type, 12 and 18 inch sizes.

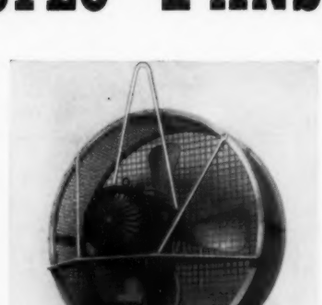
Keep Workmen Cool
with
TRUFLO FANS

During even the coldest weather there are "hot spots" in many mills—places where heat and foul air play hob with production.

Truflo Fans can keep your men on their toes all year around. All Truflo Fans are designed and built to put fresh, cool air exactly where it is needed. Perfectly balanced blades give more air per h.p. . . . strong steel frames stand up longer under hard use . . . rugged wire guards protect men against injuries.

There is a type of Truflo Fan for every cooling and ventilating use. Write for literature on any of the following kinds:

PORTABLE COOLING FANS
CRANE CAB FANS • WALL FANS
EXHAUST FANS • BLOWERS
ROOF VENTILATING FANS
PENT HOUSE FANS



TRUFLO PORTABLE
COOLING FANS
Easily portable. Help keep efficiency high where work is hottest. 12 to 36 inch diameters.

TRUFLO FAN COMPANY
554 MAIN ST., HARMONY, PA.

NEW ELECTRIC FURNACE FOR MANGANESE STEEL

▲ On October 29, a new Pittsburgh Lectromelt furnace for the production of manganese steel, was put into operation at the High Bridge, New Jersey, plant of the Taylor-Wharton Iron and Steel Company. With this installation, which appropriately enough took place on the forty-ninth anniversary of the first heat of Hadfield's 12 per cent manganese steel in America, the High Bridge plant is now on a 100 per cent electric steel-making basis.

The new furnace and equipment has been installed at a cost of about $50,000 and will be an important factor in conserving ferro-manganese in the manufacture of Hadfield's manganese steel, which is the major type of steel produced at this plant. The modern Lectromelt furnace climaxes the great progress made in the production of manganese steel since Sir Robert A. Hadfield produced the first heat in 1892, in a three-ton bottom blow converter, at this same Taylor-Wharton High Bridge plant.

At the ceremony which marked the starting of the furnace, M. F. Apgar, formerly superintendent of the High Bridge plant, and the company's present-day ranking service employee, turned the switch for the first heat. Among the officials of the company who were present were G. R. Hanks, president; L. N. Aller, secretary and treasurer; L. E. MacFadyen, superintendent of the High Bridge plant; E. E. Alexander, plant engineer. The plant C.I.O. union was represented by its president, Stanley Oleniacz.

LITERATURE AVAILABLE

▲ A new booklet entitled "Nitrogen in Chromium Alloy Steels" has just been published by Electro Metallurgical Company, a unit of Union Carbide and Carbon Corporation. This booklet shows, by the results of a large number of tests, how the mechanical properties of high-chromium steels are markedly improved by additions of nitrogen to supplement the amount of this element normally present in all steels.

The first part of this booklet discusses the effects of nitrogen in the

martensitic 12 to 14 per cent chromium steels, cutlery stainless steels, and 16-1 chromium nickel steels. It is shown that, in these heat-treatable steels, the presence of nitrogen in more than normal amounts results in improvements in the mechanical properties which could not be obtained by heat-treating alone.

Shown next are the effects of nitrogen in the ferritic non-hardenable 16 to 18 per cent and 20 to 30 per cent chromium wrought steels, the 20 to 30 per cent chromium cast steels, and nickel high-chromium wrought steels. These steels are little affected by heat-treatment but tests show that nitrogen improves their mechanical properties, especially strength and toughness.

The effects of nitrogen in the austenitic chromium steels including the well-known 18-8 stainless steels and the cast 25-12 type stainless steels are also discussed. Results of tests indicate that added nitrogen definitely increases the tensile strength of the 18-8 stainless steels, and in the 25-12 type steels nitrogen additions help maintain ductility at room temperature when these steels have been subjected to long exposures at elevated temperatures. The proper procedures, and certain precautionary methods, for adding nitrogen to chromium steels are pointed out in the latter part of the booklet.

A copy of this booklet can be obtained without charge by writing to Electro Metallurgical Company, 30 East 42nd Street, New York, New York.

▲ "Why Farval?" is an 8-page bulletin, 8½ x 11, in three colors, describing in minute detail the Farval centralized system of lubrication.

Beginning with a crisp comment on correct lubrication as a prime element in maintaining high production schedules, the booklet deals with the simplicity of the Farval system, illustrates its component parts, and charts the 12 distinct savings effected by the Farval system.

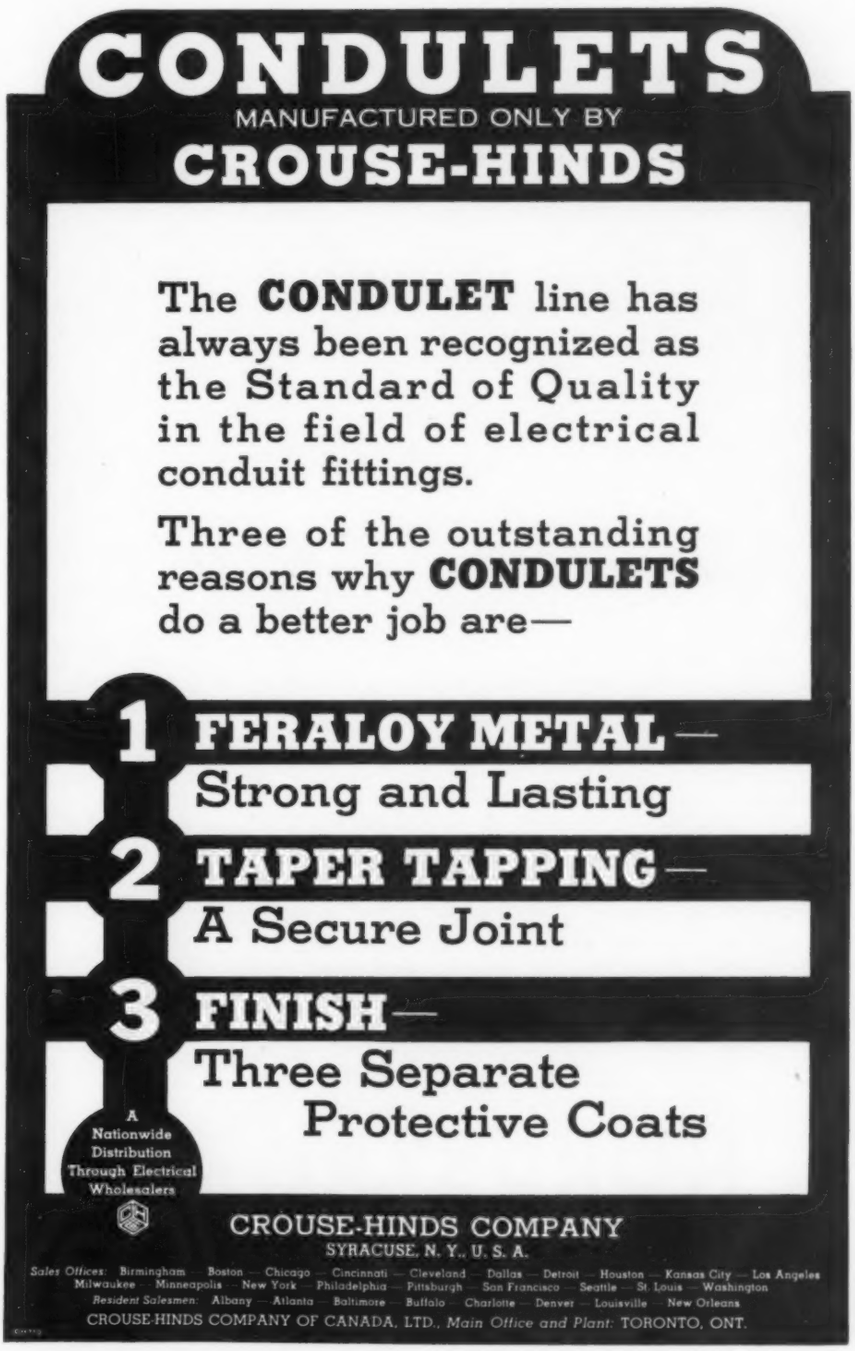
In the center spread, a series of cross-sections show the progressive movements within the Farval Dualine measuring valve as it delivers oil or grease under pressure to each individual bearing.

The new booklet is available through The Farval Corporation, Cleveland, Ohio.

▲ Allis-Chalmers Manufacturing Company, Milwaukee, Wisconsin, has recently issued a comprehensive 36-page steel mill reference book, B6197. This is a collection of articles currently written on modern rolling mill practice; control of tin plate temper; modern cold rolling of steel; variable voltage versus motor field speed control for d-c. drives; operation and maintenance of d-c. machines; and d-c. flashover and bus short circuit protection.

▲ "Background" is the title of a colorful, 8-page booklet, 8½ x 11, by the Cleveland Worm and Gear Company, Cleveland.

It emphasizes the company's claim for long-range, economical operation of Cleveland Worm Gear Speed Reducers by quoting a series of letters from manufacturers in various leading industries who began purchasing Cleveland drives 15 to 20 years ago.


CONDULETS
MANUFACTURED ONLY BY
CROUSE-HINDS
The CONDULET line has always been recognized as the Standard of Quality in the field of electrical conduit fittings.
Three of the outstanding reasons why CONDULETS do a better job are—
1 FERALOY METAL— Strong and Lasting
2 TAPER TAPPING— A Secure Joint
3 FINISH— Three Separate Protective Coats
A Nationwide Distribution Through Electrical Wholesalers
CROUSE-HINDS COMPANY
SYRACUSE, N. Y., U. S. A.
Sales Offices: Birmingham — Boston — Chicago — Cincinnati — Cleveland — Dallas — Detroit — Houston — Kansas City — Los Angeles — Milwaukee — Minneapolis — New York — Philadelphia — Pittsburgh — San Francisco — Seattle — St. Louis — Washington
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CROUSE-HINDS COMPANY OF CANADA, LTD., Main Office and Plant: TORONTO, ONT.

The various reports describe graphically the conditions under which worm gear driven machinery must operate in each particular industry. A final chapter, "Foreground," carries the message into future years of industrial progress.

▲ Catalog No. 42, covering specifications and prices of standard Kennametal steel-cutting carbide tools and blanks, has just been issued by McKenna Metals Company, 145 Lloyd Avenue, Latrobe, Pennsylvania.

The new catalog is believed to be the most complete carbide tool catalog and manual published to date. Typical applications of each style of tool are illustrated by line drawings, and photographs and information on specific examples of the use of Kennametal have been included. Several pages are devoted to selection and design of tools and blanks, correct grinding procedure, chip breaker designs, brazing tool blanks to shanks, boring tool set-ups, and other pertinent engineering data. The new Kennametal round shank boring tools, cut-off tools, roller type turning tools, and solid round tools are illustrated and described.

Copies of the new Kennametal Catalog No. 42 will be furnished on request.

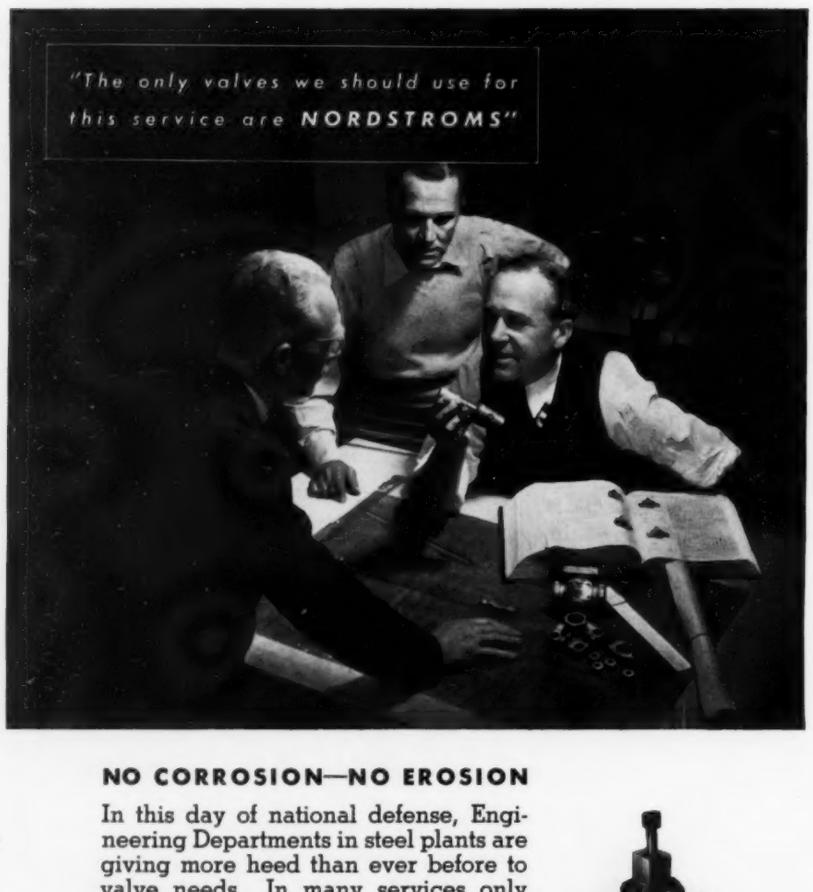
▲ The properties of refractory insulating concrete are described in a new bulletin which gives detailed information on mixing and placing this cold-setting, monolithic refractory material.

For these special types of refractory concrete light-weight insulating aggregates are used. A calcium-aluminate cement, Lumnite, is employed as the binder. The booklet stresses the availability of materials and the adaptability of refractory insulating concrete for industrial furnace construction. Six legible charts give hot- and cold-face temperatures and heat loss for different thicknesses of several types of furnace walls at operating temperatures from 250 to 2500 degrees F. Please address the Atlas Lumnite Cement Company, Chrysler Building, New York, for copies of this bulletin.

▲ A complete new catalogue describing Metaline Oilless Bronze Bearings has just been issued by the R. W. Rhodes Metaline Company, of Long Island City, New York. It is arranged in three sections. One describes the lubricating properties of Metaline; the second part outlines the various types and the range of application for oilless bearings and lastly, the catalogue presents detailed data for use in preparing specifications. Copies may be had on request.

A. I. S. E. MEMBERS AID IN HANDBOOK

▲ The seventh edition of the "Standard Handbook for Electrical Engineers," published by the McGraw-Hill Book Company, is now available. Of somewhat larger size and typography than previous editions, the book's 2303 pages present an orderly compilation of the working information of electrical technology not generally accessible in compact form. Abstract principles and theory have been condensed without sacrifice of completeness. New phases of activity and shifts in importance of older topics have been recognized by addi-



NO CORROSION—NO EROSION

In this day of national defense, Engineering Departments in steel plants are giving more heed than ever before to valve needs. In many services only Nordstrom Lubricated Plug Valves will perform to the full satisfaction of engineers. The longer life of Nordstroms, their freedom from sticking, and their resistance to corrosion and erosion, present economy advantages as well as assurance of safety. Especially desirable on gas lines. Ask for Bulletins.

KEEP UPKEEP DOWN WITH

NORDSTROM LUBRICATED VALVES

MERCO NORDSTROM VALVE CO.

400 LEXINGTON AVENUE PITTSBURGH, PENNA.

tions, expansions, and new treatments in this complete revision. Maximum space is devoted to presentation, by practicing specialists, of accepted data from the field of electrical engineering. The book was prepared under the direction of A. E. Knowlton, editor-in-chief. Among the 102 contributors were the following members of the Association of Iron and Steel Engineers: Wilfred Sykes, W. A. Perry, P. B. Harwood, and H. S. Richardson.

ANNOUNCES NEW UNITS FOR EMERGENCY POWER

▲ Meeting the increased need created by national defense activities for emergency light and power protection, the Electric Storage Battery Company, Philadelphia, Pennsylvania, announces four new Exide emergency light and power units for industrial use. The new units are completely described in a folder which can be obtained by addressing the manufacturer at 19th Street and Allegheny Avenue, Philadelphia.

The greatly increased amount of night shift production of war material has sharply emphasized the need for an absolutely reliable source of current instantly available during interruptions, unavoidable or caused by sabotage, of the regular a-c. supply at vitally important points, particularly in the engine room and at important control panels. Another important purpose of the units is to insure continuity of electric power in those delicate manufacturing processes where even a moment's interruption would result in expense and serious loss of time.

Many thousands of Exide units, embodying the same fundamental principles are in use throughout the United States and Canada, and a large number manufactured by an affiliated company in England are being used for emergency lighting in bomb-areas and in air-raid shelters—some of the latter being in famous and historic buildings.

Split-second operation of the Exide unit and absolute dependability—the two basic requirements for an emergency light and power system—are stressed by the manufacturer. These two features of the unit are made possible by the design of the unit, particularly of the transfer switch and the long-wearing quality of the material used in construction. Many years of experience by the manufacturer in making complete emergency lighting systems is pointed to as the source of the ideas and principles utilized in designing the unit.

Each unit is used in connection with an Exide battery of either the "chloride" or the "flat plate" type. The units have an automatic transfer switch which instantaneously transfers the battery to the emergency lighting circuit upon failure of the normal a-c. supply. When the a-c. service is restored, the emergency lighting circuits are instantly transferred back to the a-c. supply.

The battery is kept fully charged at all times, because immediately after normal power is restored, the battery is automatically placed on high rate charge. As soon as the recharging is completed, the high rate is automatically cut off and a copper oxide rectifier then supplies a trickle, or low rate of charge, just sufficient to maintain the battery in a fully charged condition, ready for the next emergency.

TRADE AUTOMATIC MARK

MANUFACTURERS FOR OVER THIRTY YEARS

MODERN Electric Propelled INDUSTRIAL TRUCKS

FOR ECONOMICAL MATERIALS HANDLING

- COIL AND FORK TRACTORS
- SHEET AND TIN PLATE HANDLERS
- BOSH TRUCKS
- LOW AND HIGH LIFT TRUCKS
- LOAD CARRIERS
- TRACTORS — CRANES

COIL TRACTORS
Capacities 8,000 to 18,000 lbs.

HEAVY DUTY COIL TRACTORS
Capacities up to 30,000 lbs.

FORK TRUCKS
Telescopic and Non-Telescopic Lifts
For Sheet Coils and Pallet Handling
Capacities 3,000 to 10,000 lbs.

HIGH LIFT TRUCKS
Tiering Platform Types

TIN PLATE HANDLERS
Non-Telescopic and Telescopic Lifts

REQUEST DETAILS AND LITERATURE

Listed Under Reexamination Service of UNDERWRITERS' LABORATORIES

AUTOMATIC TRANSPORTATION CO.
Div. of the Yale & Towne Mfg. Co.
47 W. 87th St. CHICAGO, ILL.

The new units, which are available in capacities of from 3400 to 10,000 watts, are particularly adapted to industrial plants engaged in defense work where interruption of the normal source of power may result in untold harm or damage. Other systems are available in capacities ranging from 240 to 100,000 watts, including those for safeguarding the continuity of highly important manufacturing processes during interruptions in the normal a-c. supply.

PLANT TO RECEIVE SHELL CONTRACT

▲ The Mansfield plant of the Westinghouse Electric and Manufacturing Company will receive a $2,750,000 U. S. Army contract for 37 millimeter armor-piercing shells, B. W. Clark, Westinghouse vice president, announced. Notification of the award was received from Army Ordnance Headquarters in Cleveland.

The contract will provide employment for several hundred persons, and machinery and other equipment necessary for this work will be ordered immediately. Production should be started shortly after the first of the year.

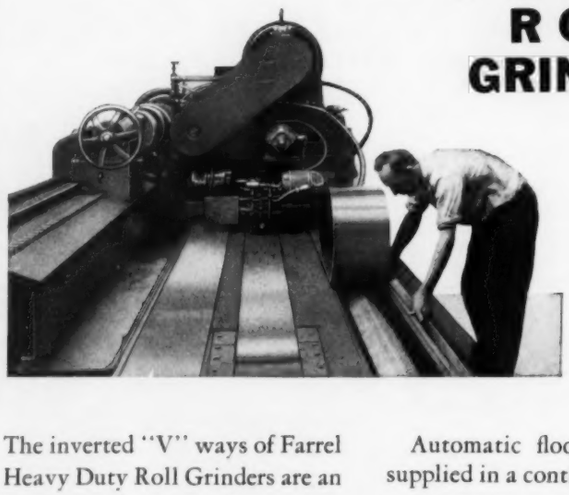
In order to speed this project in every way possible, the men who will act as supervisors in the shell-producing section are being sent to government arsenals to familiarize themselves with the work.

The 37 millimeter shells, which are approximately 1½ in. in diameter and 6½ in. in length, are particularly effective in anti-tank warfare. Known technically as three-piece shot, they have a forged steel body with an alloy steel penetrating point or cap, and an aluminum "wind-shield" which cuts down wind resistance.

Westinghouse officials some months ago spent $75,000 to prepare a section of the Mansfield plant to handle work of this type and the new contract will be their second major defense project. In August the plant was awarded a $3,173,000 order for the manufacturing of Army binoculars. More than half the total production of Westinghouse plants throughout the country is devoted to defense work.

INVERTED "V" WAYS *Contribute to High Precision and Productive Capacity of*

FARREL ROLL GRINDERS



The inverted "V" ways of Farrel Heavy Duty Roll Grinders are an important factor in the smooth, accurate, uniform traverse of the carriage, so necessary for precise roll contour and high quality finish.

Cast separately from the bed and made of a special alloy iron of unusual hardness and close grain, these inverted "V" ways are exceptionally resistant to wear. In addition, the ways are made with a large area of bearing surface which makes the unit pressure low, still further retarding wear and lengthening their accurate life.

Automatic flood lubrication, supplied in a continuous cycle by a separate motor-driven oil pump, cleanses them of foreign matter which might settle on them. Double protection is afforded by the flexible metal guards which fit closely over the way channels to exclude dirt and grit.

The inverted "V" ways are but one of the many features of Farrel Heavy Duty Roll Grinders that contribute to the high accuracy and high productive capacity of these precision machines. Full details are given in our Bulletin No. 111, which will be sent free on request.

FARREL FARREL-BIRMINGHAM COMPANY, Inc.
ANSONIA, CONN.
New York • Buffalo • Pittsburgh • Akron • Chicago • Los Angeles

BOOKLET DISCUSSES WATER TREATMENT

▲ The characteristics of modern boilers have placed complex water treatment problems on plant operators. The higher pressures and larger sizes of boilers now in common use result in fewer spare boilers available in case of trouble, a condition demanding reliable water conditioning. The maintenance of a protective film, control of alkalinity, and the prevention of embrittlement and deposits in the boiler are fully discussed in this new 8-page leaflet—a valuable aid to everyone interested in reliable water conditioning. Address Hall Laboratories, Inc., 300 Ross Street, Pittsburgh, Pennsylvania.

RAPID METHOD FOR SPECTRUM ANAYLSIS

▲ A rapid and convenient method for analyzing spectrographic plates or films in research and industrial laboratories is provided by a new recording microphotometer recently announced by Leeds and Northrup Company, 4934 Stenton Avenue, Philadelphia, Pennsylvania. Because the plates or films are mechanically scanned by a

motor-driven scanning unit, and the relative positions and densities of spectrum lines are automatically recorded by an L & N speedomax recorder, this new recording microphotometer eliminates the eyestrain and "human equation" of visual methods, and does away with the delays and limitations of photographic processes. Quick determinations of elements present in small percentages are made possible by pen-and-ink records of standard and test spectrograms drawn at high speed on one chart.

Designed to accommodate plates or films as large as 4 in. high by 10 in. wide, the scanning unit includes on a heavy cast base, an optical system, a plate stage, a drive mechanism for the plate stage, an a-c. operated amplifier and all necessary controls grouped at a convenient location. In indicating and recording relative positions and densities of spectrum lines, the response of the speedomax recorder to density changes is so immediate and so rapid that in less than 1.5 seconds the pen moves from one position and comes to rest at its new position . . . even across the entire width of the 9 in. chart. Pen position can be read at a glance on an indicating scale as the record is being drawn. No unusual skills are required to operate this new microphotometer. Anyone familiar in a general way with instruments can quickly learn to use it.

RESPIRATOR OFFERS DUST PROTECTION

▲ A new respirator, which affords maximum protection against the inhalation of pneumoconiosis-producing and nuisance dusts, has just been announced by the DeVilbiss Company, Toledo, Ohio.

Approved by the United States Bureau of Mines, the new DeVilbiss "MSD" respirator is said to embody nine unique features. These combined, make it more compact, lighter, easier to breathe through, more comfortable to wear, and easier to clean and maintain than previous models.

In the new respirator, a scientifically and compactly designed cartridge, measuring only 2⅞ by 1½ in. and weighing but one ounce, actually provides more than 41 sq. in. of breathing and filtering area. Ingenious multi-vane construction allows every inch of this filter to be used efficiently. This means no air restriction to cause difficulty in breathing, plus greatly increased filtering efficiency.

The filter is easily cleaned by blowing accumulated dust from vanes with an air dusting gun or nozzle, thus maintaining peak performance and requiring less frequent replacement.

The new DeVilbiss "MSD" respirator is designed to fit, closely and without leakage, the contour of practically any face. This is an important feature, for full protection depends upon a good fit.

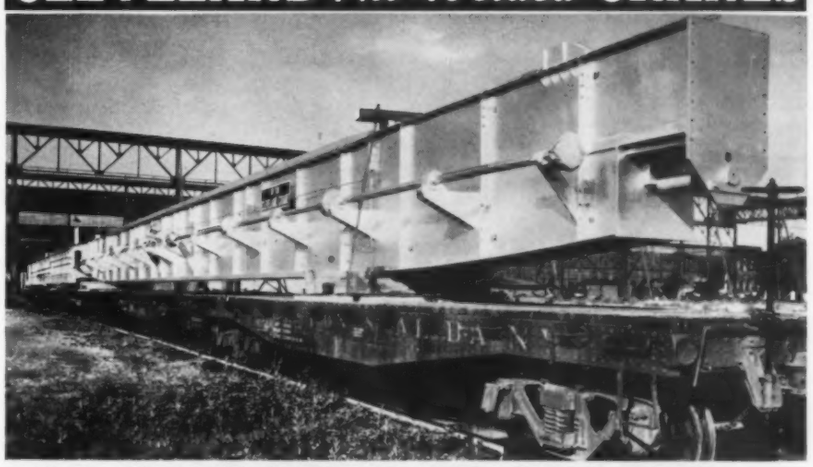
WATER SOFTENING BOOK

▲ Cochrane Corporation has on hand copies of an attractively bound book on Cochrane Hot Process Softeners. The processes and apparatus used in water softening are described and illustrated in a thorough manner.

Steam plant engineers should write to the Cochrane Corporation, 17th Street and Allegheny Avenue, Philadelphia, Pennsylvania.

15TH ANNIVERSARY

CLEVELAND *All-Welded* CRANES



122 CARLOADS OF CRANES

on *one* order

EVERY CRANE ALL-WELDED

That's a big order! But more important, there must be a great deal of confidence in a company and its product when an order of that magnitude is placed with it.

Cleveland Crane has the confidence of steel mill engineers. From experience with all kinds of cranes, they know that no crane can excel an all-welded crane (meaning of course, a crane welded throughout, including the girders) for *hard* day in and day out service.

That's why big orders for Cleveland all-welded Cranes are placed without hesitation.

THE CLEVELAND CRANE & ENGINEERING CO.
Wickliffe, Ohio

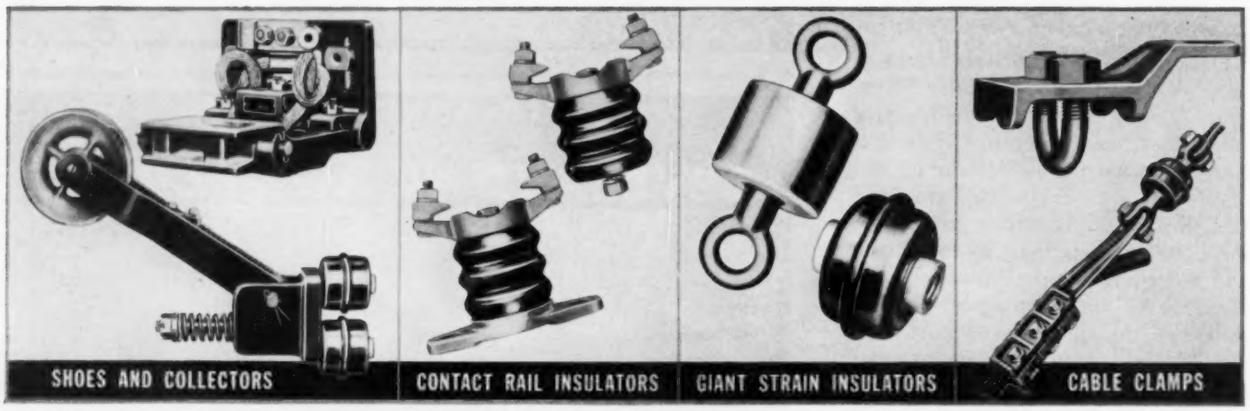
CLEVELAND CRANES

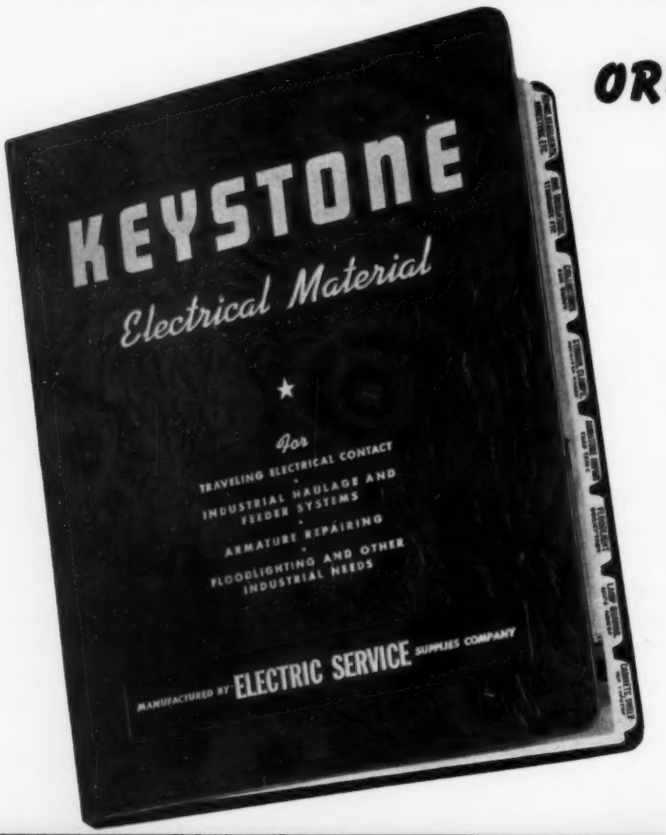
ALL-WELDED OVERHEAD TRAVELING CRANES

Other products: CLEVELAND TRAMRAIL and STEELWELD MACHINERY

Vital Material

FOR YOUR CRANES AND FEEDER SYSTEMS





ORDER SPARES NOW!

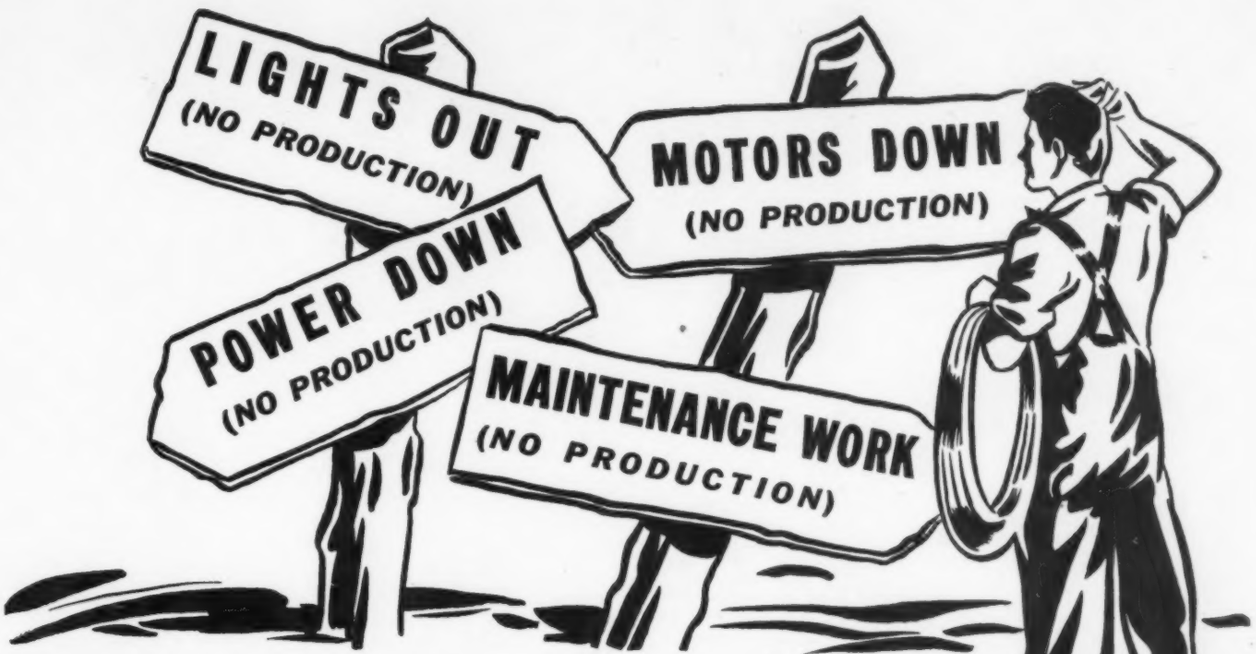
★ Contact Rail Shoes or Collectors and Insulators are vital in the operation of cranes and other haulage systems. Strain Insulators and Clamps are essential for heavy feeder systems. Orders for these materials exceed production and we suggest that you anticipate your future needs (including spares for emergencies) so that we can keep you supplied and not hold up your production.

★

CATALOG AND PRICES SENT PROMPTLY ON REQUEST

MANUFACTURED BY **ELECTRIC SERVICE** SUPPLIES COMPANY

17th & CAMBRIA STREETS • PHILADELPHIA • PA. • BRANCHES IN PRINCIPAL CITIES AND FOREIGN COUNTRIES



Why be Perplexed?

DO SOMETHING about it!

If *your* lighting, power and control circuits are quitting on you and clogging up production, and motor lead failures are knocking out machines just when you need them most . . . *don't replace the old stuff and hope* . . . rewire with *permanently insulated Rockbestos.*

Right now when every ounce of production is needed, high overloads and the continuous operation of lighting, power and control circuits are building up trouble that's bound to cause annoying failures when you least expect them *and when you can least afford them.* Copper-heating temperatures and the high ambients that attack circuits close to boilers, furnaces, drying ovens, kilns, soaking pits, steam lines, etc., dry out and crack ordinary insulation and the failure which follows kicks your production out the window.

If that's your situation we suggest that you keep a wire failure chart so that you can spot your "repeaters," calculate your *minimum requirements* and rewire with permanently insulated Rockbestos wire, cable or cord.

For a catalog or samples write to: Rockbestos Products Corporation, 821 Nicoll St., New Haven, Conn.

Anticipate Your Requirements

Also refer to McGraw-Hill Electrical Buyers Reference

Rockbestos A.V.C. Boiler Room and Lighting Wire (Underwriters' and Nat. Elec. Code—Type AVA maximum operating temperature 110°C. (230°F.)

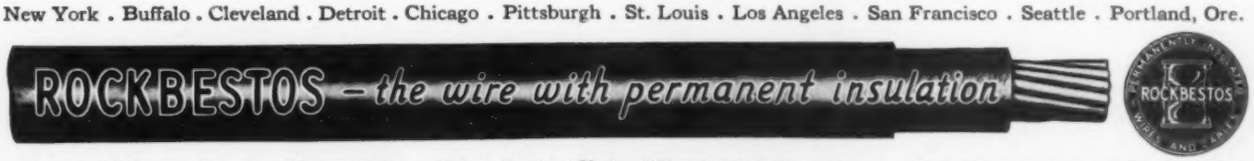
Rockbestos A.V.C. 600 Volt Power Cable (Underwriters' and Nat. Elec. Code—Type AVA maximum operating temperature 110°C. (230°F.)

Rockbestos A.V.C. 600 Volt Motor Lead Cable (Underwriters' and Nat. Elec. Code—Type AVA maximum operating temperature 110°C. (230°F.)

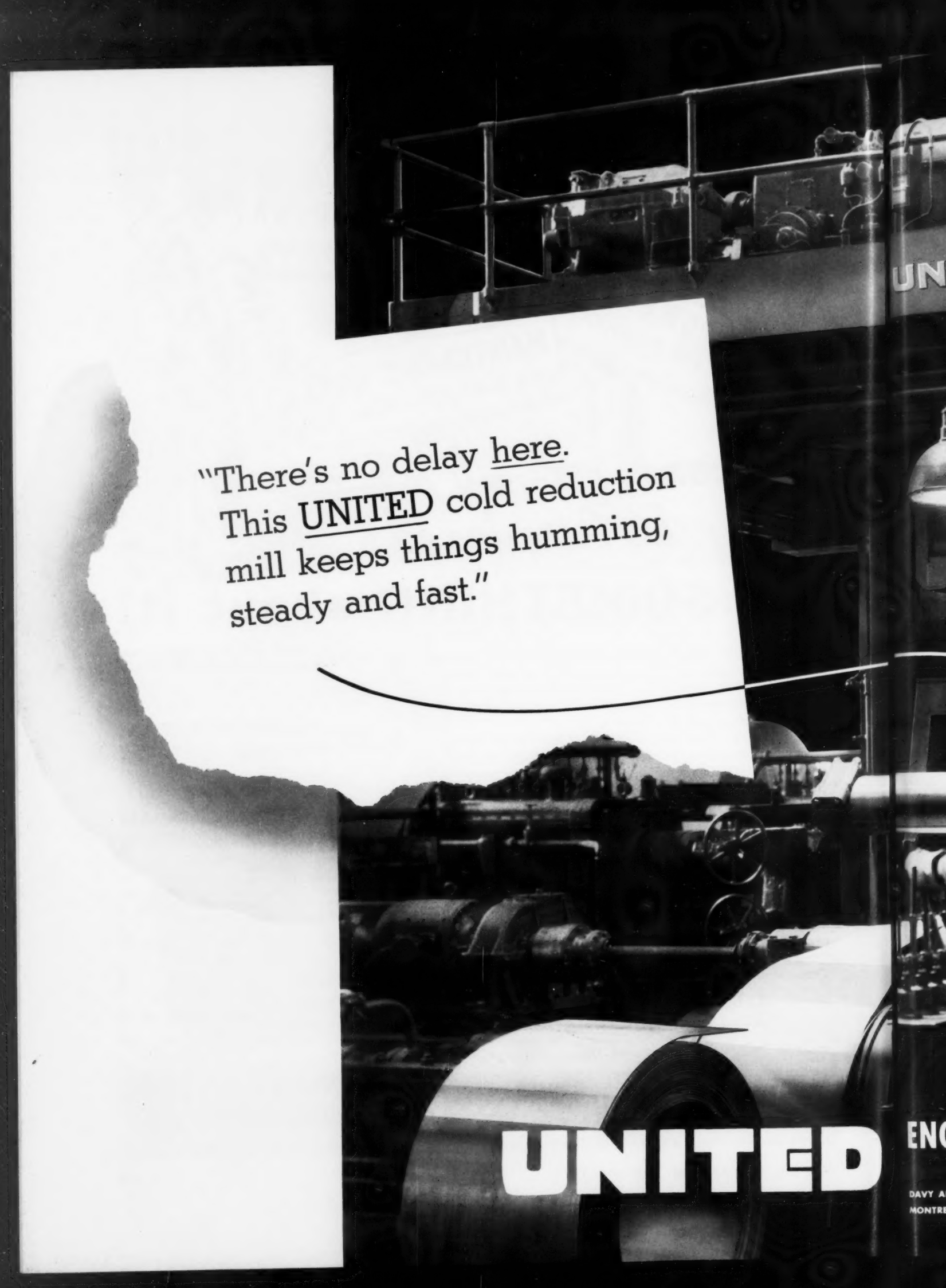
TEN TESTED ROCKBESTOS VALUES that will pay you dividends

1. HEATPROOF
2. FIREPROOF
3. PERMANENT
4. Lower maintenance cost
5. Resists heat and vibration
6. Saves work
7. Oil, Grease and Moisture resistant
8. High overload capacity
9. Permanently flexible
10. Greater carrying capacity

New York . Buffalo . Cleveland . Detroit . Chicago . Pittsburgh . St. Louis . Los Angeles . San Francisco . Seattle . Portland, Ore.



Rockbestos All-Asbestos Power and Rheostat Cable — Underwriters' and National Electrical Code Type AI — one of 118 different permanently insulated wires


"There's no delay here.
This UNITED cold reduction
mill keeps things humming,
steady and fast."
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ENGINEERING *and* FOUNDRY COMPANY

Pittsburgh, Pennsylvania

DAVY AND UNITED ENGINEERING COMPANY, LTD., SHEFFIELD, ENGLAND · DOMINION ENGINEERING WORKS, LTD., MONTREAL, P. Q. · UNITED INTERNATIONAL, S. A. PARIS, FRANCE · SHIBAURA-UNITED ENGINEERING CO., TOKYO, JAPAN

1901 U E & F D FORTY YEARS SERVICE 1941

ITEMS OF INTEREST

Robert H. Bahney has been promoted from the position of chief engineer of the Central Alloy District of Republic Steel Corporation, with headquarters in Canton, to assistant chief engineer for the corporation, with headquarters in Cleveland. Mr. Bahney is succeeded by **Eugene W. Phillips,** who has been his assistant.

Mr. Bahney, who was born in Massillon, started as a wireman's helper in the line gang with the former Central Steel Company in 1915. After completing his college course at Cornell University he returned to the company as clerk in the electrical office in 1917, and in 1919 was made chief electrician. When the Central Alloy Steel Corporation was formed in 1926, through the merger of Central Steel Company, Massillon, and United Alloy, Canton, Mr. Bahney was made chief electrician of the Canton properties. Five years later, when Central Alloy became a part of the newly organized Republic Steel Corporation, he was promoted to assistant superintendent of the mechanical and electrical departments in Canton, and in 1936 went into the engineering department. In 1939 he was promoted to the position which he just left, of chief engineer of the Central Alloy District.

Mr. Phillips attended school in Canton, graduating from Canton High School in 1915. His work at Marquette University, Milwaukee, from which he graduated in 1919 with the degree of B.S. in M.E., was interrupted by several months in the Engineering Corps of the Army during 1918.

In 1919 he became associated with the United Furnace Company in Canton, and in 1921 joined the United Alloy Company, where he was field engineer, later becoming superintendent of construction. In 1932 he was transferred to the industrial engineering department of Republic's Youngstown District, and returned to Canton in 1935 as field engineer.

Clarence R. Burrell has been appointed general sales manager of the Tool Steel Gear and Pinion Company, Cincinnati, Ohio. Mr. Burrell was graduated from the University of Illinois in 1927, and worked at Inland Steel Company for two years after graduation. His next employment was with Lubrication Devices, Inc., serving with them as a salesman in Chicago, as a distributor in Pittsburgh, and as chief engineer at their factory. When this company was taken over in 1932 by the Cleveland Worm and Gear Company, and the name changed to Farval Corporation, Mr. Burrell served as an assistant to the sales manager until 1937. At that time the sales organizations of the Worm Gear Company and the Farval Corporation were combined, Mr. Burrell becoming assistant sales manager of both companies, in which position he remained until joining the Tool Steel Gear and Pinion Company in April of this year in a sales capacity.

R. Russell Fayles, formerly engineer, South Chester Tube Company, Chester, Pennsylvania, has joined Lukens Steel Company, Coatesville, Pennsylvania, as combustion engineer in the open hearth department.

J. V. Freeman has been appointed assistant to the vice president in charge of coke by-product sales of all subsidiary companies of the United States Steel Corporation of Delaware. His office is at 71 Broadway, New York. Mr. Freeman has been with the United States Steel Corporation and its subsidiaries since 1908, beginning as coke inspector at Gary, West Virginia, and later was transferred to Joliet, Illinois, during the construc-

ROBERT H. BAHNEY
"asst. chief engineer all Republic plants"

CLARENCE R. BURRELL
"will head sales for Tool Steel Gear & Pinion Co."

tion of the Joliet by-product coke ovens, Illinois Steel Company. In 1910 he became chief chemist, central coal research laboratory, Joliet, where he remained until 1925, at which time he was transferred to the New York offices of the corporation as assistant in the technical and by-product sales department.

K. C. Stevens has been elected as president of the Pittsburgh and Conneaut Dock Company, Conneaut, Ohio, and president of the Pennsylvania and Lake Erie Dock Company, Fairport, Ohio, United States Steel Corporation subsidiaries, as announced by the Boards of Directors of the two companies. Mr. Stevens succeeds **George S. Meek,** who died last year. Mr. Stevens was graduated from Marietta College in 1925 and the Case School of Applied Science in 1928 with a B.S. degree in mechanical engineering. Following his graduation, he went with the Marietta Concrete Corporation as a mechanical engineer.

In 1929 he became associated with the Pittsburgh and Conneaut Dock Company as assistant master mechanic. He then became successively assistant to general superintendent, vice president and superintendent of the Pittsburgh and Conneaut Dock Company and assistant to the president and manager of the Pennsylvania and Lake Erie Dock Company.

Clyde L. Ross was elected vice president and will continue as auditor and assistant secretary. **George H. Bruce** was appointed assistant to general superintendent.

Robert Carter has been appointed purchasing agent of the Pittsburgh Limestone Company, a subsidiary of the United States Steel Corporation. Mr. Carter is a native of Pittsburgh, and was previously in the purchasing department of American Sheet and Tin Plate Company and of Carnegie-Illinois Steel Corporation.

ROBERT CARTER
"new Purchasing Agent, Pittsburgh Limestone Co."

CONGRATULATIONS MR. JENKS . . . . .

S. M. Jenks, general superintendent of Gary Works, Carnegie-Illinois Steel Corporation, receives congratulations from S. M. Havens, vice president, Greater Chicago Safety Council, for Gary Works' part in winning the 25th semi-annual Inter-plant Accident Prevention Contest in the first six months of this year. Mr. Jenks is holding the bronze plaque awarded by the Council to the Gary Works and Joliet Coke plant for having the best safety record among industrial plants in the Chicago-Gary area, recording 5,000,000 man-hours of work or more.

H. G. Hilton, vice president of the Steel Company of Canada, Limited, has been elected a director.

Mr. Hilton is a graduate of Case School of Applied Science and before his association with the Steel Company of Canada, Limited, was connected with the blast furnace operations of Pickands Mather and Company.

He came to the Steel Company of Canada, Limited, in 1919 as blast furnace superintendent, became assistant works manager in 1927 and in 1934 was made works manager. Mr. Hilton was appointed vice president in 1937 and his appointment to the Board of Directors of the Steel Company of Canada, Limited, fills the vacancy left by the Right Hon. Arthur B. Purvis.

Robert M. Gibson has been transferred from Koppers Company, Bartlett Hayward Division, to Koppers Company, American Hammered Piston Ring Division, Baltimore, as engineer. He is engaged in design and construction to greatly increase the facilities there.

"Spike" Gibson, as he is perhaps better known, became connected with the blast furnace and steel plant industry after his return from England, where he served in the A.E.F. during the World War. He joined Koppers Company in 1923, was assistant district engineer in the New York office and later assistant superintendent of maintenance for the Seaboard division. Since 1937 he was sales engineer of equipment for blast furnace, steel plant and process industries for the Bartlett Hayward division.

A. F. Giese, recently promoted to division superintendent of maintenance of the Gary Works, Carnegie-Illinois Steel Corporation, succeeds **T. R. Miller** who was transferred to the Chicago District. Mr. Giese was employed as general foreman in the 76 in. hot strip mill at Inland Steel Company until 1937, when he went with Carnegie-Illinois as superintendent of electrical and mechanical maintenance at the coke plant. He successively held the positions of chief designing engineer in the engineering department, superintendent of the 36 in. slab mill, and superintendent of the 44 in. blooming mills.

Charles H. Armstrong, since 1936 a sales engineer in the Detroit office of the Clark Controller Company, Cleveland, has been appointed assistant district manager of the Michigan sales territory.

Graduated as an electrical engineer from the University of Michigan in 1935, he spent a year in the plant and office of the Clark Controller Company, then came to Detroit where he is well known in the electrical control field.

W. B. Simons, Charlotte, North Carolina, has been appointed representative for Cochrane flow meters in the Carolinas, according to announcement from Cochrane Corporation, Philadelphia. Mr. Simons also handles the products of Hays Corporation, Spence Engineering, Cuno Engineering, Williams Gauge, and the Henszey Company.

Norman P. Goss has been appointed a research engineer, American Steel and Wire Company, Cleveland. Mr. Goss formerly was with the Cold Metal Process Company, Youngstown, Ohio.

Clem W. Gottschalk, since 1932 assistant traffic manager, Jones and Laughlin Steel Corporation, Pittsburgh, has been made general traffic manager, succeeding **H. E. Graham,** who retired November 1.

Robert D. Knight has been appointed superintendent of the spring mill, South Works, American Steel and Wire Company, Worcester, Massachusetts. He formerly was assistant general foreman in the spring mill. **Harry F. Clarke** has been made assistant superintendent of the same plant, while **George E. Hansen** has been named product engineer in the metallurgical department at South Works.

The November meeting of the A.I.S.E. Pittsburgh District Section, on the subject of increased production from existing equipment, was attended by 400 steel plant engineers and operators.

The program covered coke plants, blast furnaces, open hearths, bessemer and duplexing, and mill equipment. The speakers were *(left to right)* F. C. Swartz, open hearth superintendent, Carnegie-Illinois Steel Corporation, Youngstown, Ohio; L. E. Riddle, blast furnace superintendent, Carnegie-Illinois Steel Corporation, Duquesne, Pennsylvania; W. T. Brown, research engineer, Jones and Laughlin Steel Corporation, Pittsburgh; L. P. Lias, plant metallurgist, Jones and Laughlin, Pittsburgh; and Joseph Malborn, United Engineering and Foundry Company, Pittsburgh.

Obituaries

Floyd C. Smith, associated with the Crouse-Hinds Company, Syracuse, New York, for the past 35 years, died recently. Mr. Smith originally worked in the engineering department, later advancing to sales work. He was a familiar figure at the A.I.S.E. conventions and meetings for more than 20 years, and he will be missed by his many friends in the industry.

A. F. GIESE

CHARLES H. ARMSTRONG

FLOYD C. SMITH

MODERN BEARINGS for MODERN NEEDS

On Roll Necks, Universal Couplings, Spindle Carriers, Table Rolls, Cranes and other tough jobs the many types of GATKE Fabric Bearings offer mighty aid in eliminating delays and avoiding costly machine shop work. An installation on your toughest service will show you. Write for particulars.

Gatke MOULDED FABRIC Bearings

GATKE CORPORATION - 228 N. La Salle - CHICAGO

William Beye, vice president, United States Steel Corporation of Delaware, died at his home in the Schenley Apartments, Pittsburgh, Pennsylvania, Monday, October 27.

During his entire career, Mr. Beye was interested in the broad aspects of industrial relations and was credited with the authorship of the Illinois Workmen's Compensation Act.

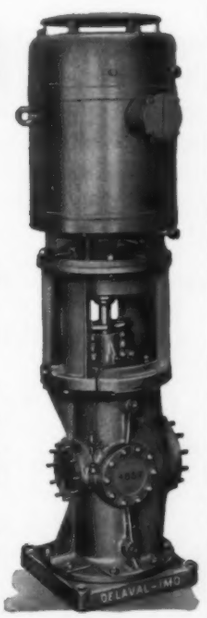
Robert H. Westerberg, superintendent of the inspection department of the Gary Works of the Carnegie-Illinois Steel Corporation, died October 8 in Gary, Indiana. Before his transfer six years ago, he was connected with the W. Dewees Wood Works in McKeesport, Pennsylvania, for 30 years. Mr. Westerberg was born in McKeesport in 1893.

Harry Dietrick, 60, a foreman of the strip mill, Republic Steel Corporation, Cleveland, died October 23 in that city. He had been with the corporation for 23 years.

Carl H. Glaser, 51, blast furnace superintendent, Carnegie-Illinois Steel Corporation's Ohio works furnaces at Youngstown, Ohio, died October 6 in a Detroit hotel. He had previously collapsed at the Ohio works due to overwork on behalf of defense orders, and had been ill since. He was born March 1, 1890, in Warren, Ohio, was graduated from Case School of Applied Science as a mechanical engineer, had been employed by Carnegie-Illinois since 1913 and went to the Youngstown plant 17 years ago from Steubenville, Ohio. In the World War he served in France as a captain in the 309th Engineers.

IMO Rotary OIL PUMPS

("IMO" means "I Move Oil")



THE unit illustrated, although occupying only a 20 in. square of deck space, delivers 500 g.p.m. of turbine lubricating oil.

IMO Oil Pumps are available in all capacities, for all pressures, and for all kinds of oil.

Catalog I-64 describes them in detail.

IMO PUMP DIVISION
of the De Laval Steam Turbine Co.
Trenton, New Jersey

WORLD'S GREATEST all around ELECTRIC TOOL

DRILLS—GRINDS—SANDS
SAWS—POLISHES
SHARPENS—CARVES

ONLY $8.95 POSTPAID
GUARANTEED FOR ONE YEAR

The new WHIZ ELECTRIC TOOL is the handiest power tool ever made. A rugged tool for power and precision work. Drills through ¼ inch iron plate in 42 seconds or engraves intricate designs. Handles any material: Metals—Woods—Alloys—Plastics—Glass—Steel—etc. Saves time. Eliminates labor. Plug into any socket AC or DC, 110 volts. Chuck ¼ inch capacity. Ball bearing thrust. Powerful, triple-geared motor. STANDARD MODEL, with Normal Speed (uses 200 different accessories, instantly interchangeable). **Price only $8.95.**

The only DRILL-TOOL with a full year's guarantee

FREE Accessory outfit (Value $2) includes set of drills, mounted 1½ inch grinder, sanding discs, cutting wheels, mounted brush, polishing wheel, carving burr, etc. FREE with each tool ordered NOW. We pay postage.

10-DAY TRIAL—MONEY BACK GUARANTEE

PARAMOUNT PRODUCTS CO.

Dept. II-RAS 545 Fifth Ave. New York, N. Y.

A SINGLE OBJECTIVE.....

Production Control

A SINGLE RESPONSIBILITY IN ACHIEVING IT

Higher tonnage and uniform quality of product result from efficient control of steel mill equipment. An invaluable aid in achieving this is a system of automatic controls and instruments that allows operators to concentrate on the factors making for optimum production. To secure these results, Republic Flow Meters Co. offers a complete manufacturing and engineering service—a single responsibility—in the field of measurement and control. We will gladly co-operate with you in the solution of any metering or control problem. Your inquiries involve no obligation.

FURNACE CONTROLS
For furnace pressure, fuel, fuel-air ratio.

BOILER CONTROLS
For all boilers, all types of firing.

REGULATORS
For pressure, flow, speed, level, ratio.

GAS MIXER CONTROLS
For mixing gases in desired ratio.

COKE OVEN GAS CONTROLS
For collecting mains, stack draft, pressures.

RING-BALANCE METERS
For steam, water, gas, air, oil, etc.

ELECTRICAL FLOW METERS
For steam, water, gas, air, oil, etc.

FURNACE PRESSURE RECORDERS
For pressure in all types of furnaces.

DRAFT & PRESSURE INDICATORS
For draft, pressure, flow, temperature, CO_2, etc.

THERMOMETERS
For temperatures up to 1000 F.

DATA BOOKS MAILED ON REQUEST

REPUBLIC FLOW METERS CO.

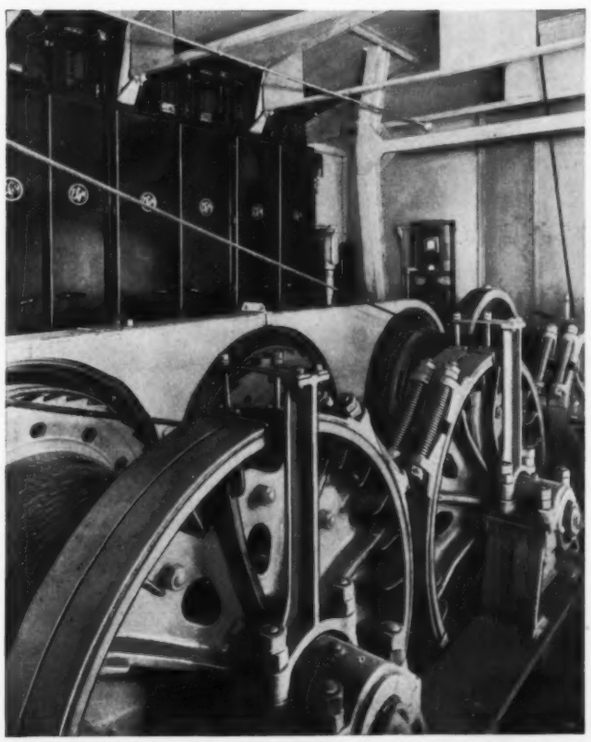
2224 Diversey Parkway, Chicago, Illinois

NO JAR • NO BUMP • NO JOLT

You can make a

CUSHIONED LANDING

When EC&M FREQUENCY RELAYS *Safeguard* A-c Motor Hoists!



Interior view of crane showing EC&M Enclosed Controllers with Nickel Alloy Resistors on top of Cabinet. Compact EC&M Master Switches permit convenient grouping at operator's station shown in small view above.

AMERICA'S Railroads have a reputation for careful handling of *all* types of shipments,—large, small, heavy or bulky. One of the large Eastern railroads has recently improved its facilities with this new portal crane which is *outstanding* in its *ability* to lower a bulky shipment slowly through the restricted opening of a hatch or rest a heavy cargo carefully in the ship's hold.

The operator can attain results with this crane which previously have been impossible with cranes powered by A-c wound-rotor motors. The reason is that EC&M Frequency Relay Magnetic Control permits *weaker values of torque*—so essential to smooth and slow lowering of light and intermediate loads. It is perfectly safe to get slow speeds by means of these weak-torque points because Frequency Relays guard against overspeeding should these points be used to lower a heavy load.

Since the introduction of EC&M Frequency Relay Control, a new perspective has been drawn relative to A-c cranes. Engineers and management have been quick to alter their ideas on crane performance after witnessing this new control in operation. Perhaps there is an installation in your vicinity that can be visited. Consult with EC&M before writing A-c crane specifications.

THE ELECTRIC CONTROLLER & MFG. CO. E&M CLEVELAND, OHIO

Specify EC&M FREQUENCY RELAY Control for A-c CRANE HOISTS

LESS COST PER TON OF STEEL ROLLED Rounds

The men who do the actual rolling in the mills can tell the story of roll performance as no one else can. Ask any mill man about Pittsburgh Rolls.

PITTSBURGH ROLLS

PITTSBURGH ROLLS DIVISION of BLAW-KNOX CO.—"Steel's Partner" • PITTSBURGH, PA.

IT
IS THE
TIME OF
YEAR FOR THE
TRADITIONAL
SENTIMENTS OF
PEACE ON EARTH AND
GOOD WILL TO MEN. THIS
YEAR, HOWEVER, WORLD
CONDITIONS MAKE SUCH
GREETINGS TRAGICALLY UNREAL.
PERHAPS THERE IS IN US THE POWER
TO SEE, FAINTLY DISCERNIBLE, IN THE
LIGHT OF THINGS TO COME, A BETTER
WORLD IN THE FUTURE. — IN EXPRESSING
OUR GOOD WISHES TO ALL OUR FRIENDS
AND CUSTOMERS, MAY WE RENEW OUR PLEDGE
TO HELP OUR COUNTRY IN PROVIDING WHAT WE
BELIEVE TO BE A BETTER AMERICA FOR US ALL!

Pennsylvania
TRANSFORMER COMPANY
808 RIDGE AVENUE N S PITTSBURGH, PA

SIMPLE, PRACTICAL FEATURES FOR BETTER WIRING

Pylets—the improved heavy duty conduit fittings—were designed for faster, easier installation, for greater strength and maximum security of wiring in heavy duty service. Consider these features: malleable bodies have double weatherproof finish; heavy hubs with accurate taper threads simplify correct installation; smooth, rounded edges, cover screws in corners, and large wiring space prevent damage to wires; reinforced, warped covers are designed to insure tight and uniform gasket seal; self-aligning covers and dowel-pin type Everdur screws are self-positioning for easiest handling.

These are some of the features that distinguish Pylets; that are bringing an ever-increasing use of these improved conduit fittings in representative mills in all parts of the country.

The complete Pylet line is listed in special bulletins which will be sent upon request. Find out what Pylet advantages can mean to you.

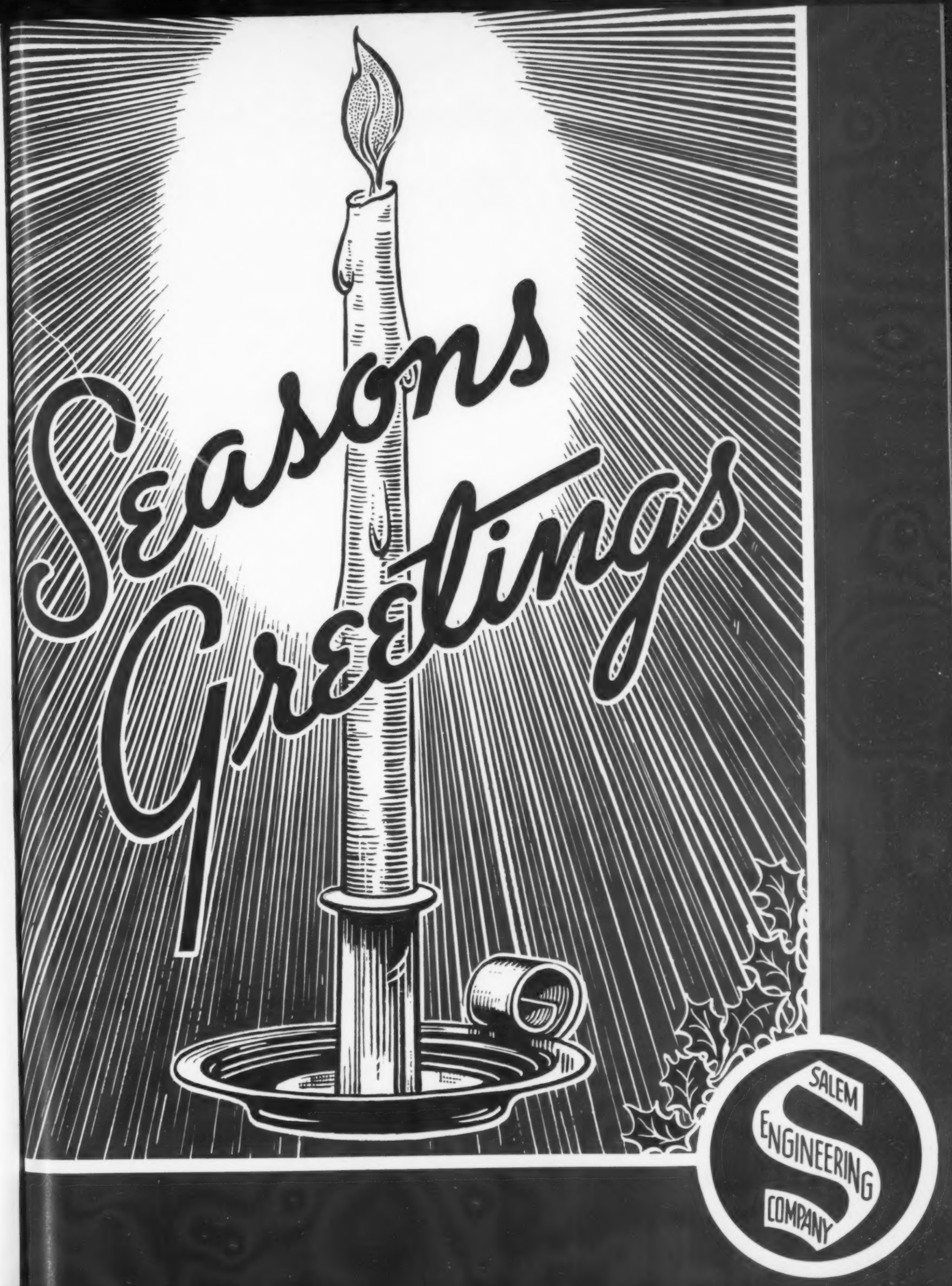
PYLE-NATIONAL
ELECTRICAL EQUIPMENT
Since 1897

THE PYLE-NATIONAL COMPANY

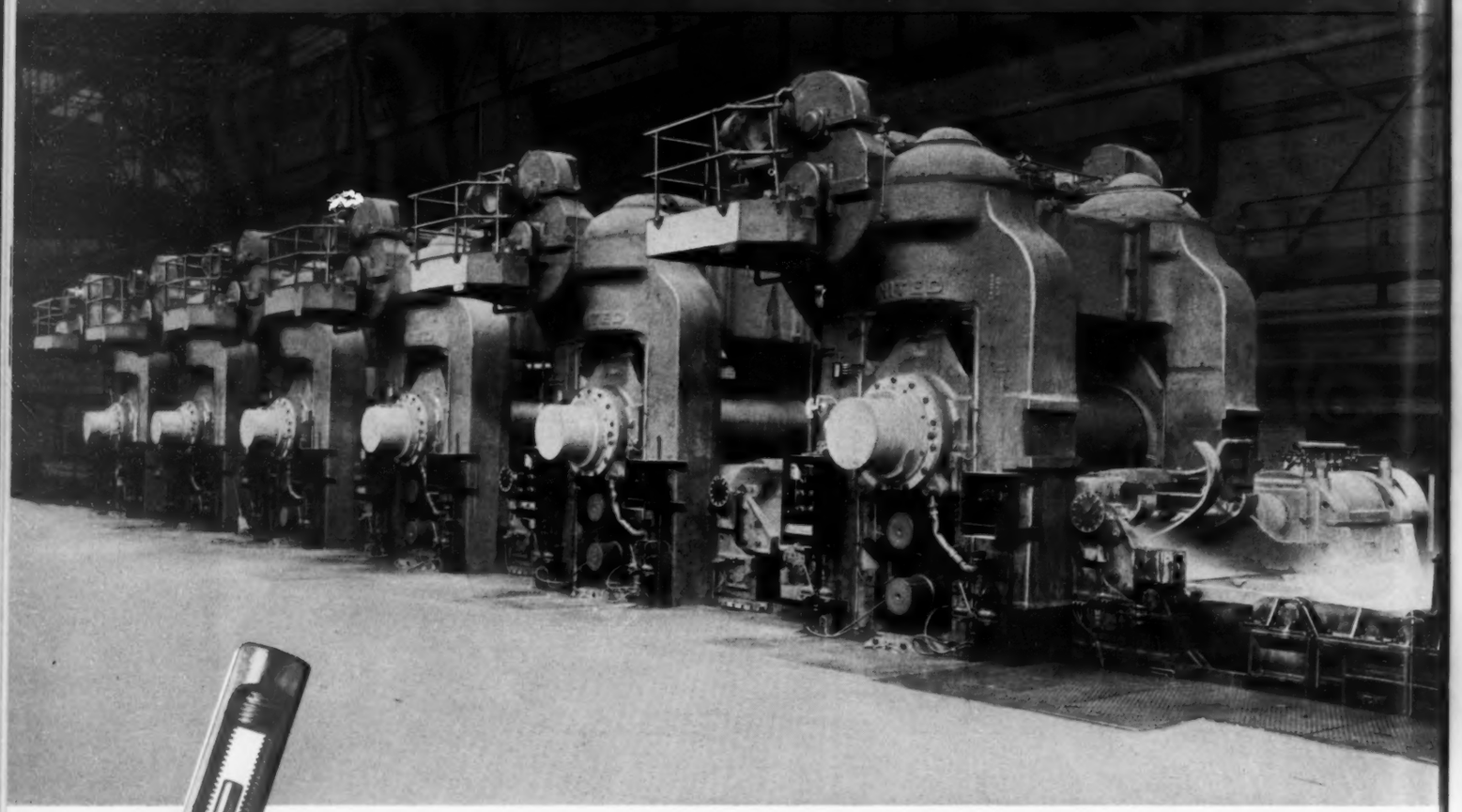
1334-58 NORTH KOSTNER AVENUE CHICAGO, ILLINOIS

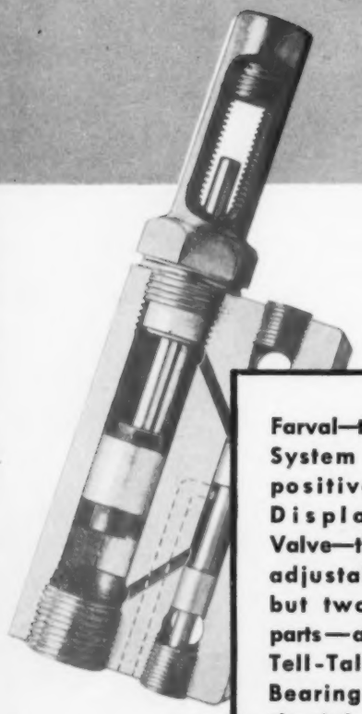
Offices: New York • Baltimore • Pittsburgh • St. Louis • St. Paul • San Francisco

CONDUIT FITTINGS • LOCOMOTIVE ELECTRICAL EQUIPMENT • FLOODLIGHT PROJECTORS


Seasons
Greetings
SALEM
ENGINEERING
COMPANY

70% of the Lubricating Systems on Hot Strip Mills are Farval





Farval—the Dualine System with the positive Piston Displacement Valve—that is fully adjustable—has but two moving parts—and with a Tell-Tale at each Bearing to show the job is done.

FARVAL HOT STRIP MILL INSTALLATIONS

CARNEGIE-ILLINOIS STEEL	38" Mill	REPUBLIC STEEL	98" Mill
CARNEGIE-ILLINOIS STEEL	80" Mill	WHEELING STEEL	66" Mill
CARNEGIE-ILLINOIS STEEL	96" Mill	YOUNGSTOWN SHEET & TUBE	79" Mill
GRANITE CITY STEEL	90" Mill	YOUNGSTOWN SHEET & TUBE	54" Mill
INLAND STEEL	76" Mill	JOHN SUMMERS & SONS, LTD.	60" Mill
INLAND STEEL	44" Mill	MACHINOIMPORT, *Russia*	32" Mill
JONES & LAUGHLIN STEEL	96" Mill	NIPPON SEITETSU, *Japan*	86" Mill
OTIS STEEL	77" Mill	NIPPON SEITETSU, *Japan*	43" Mill

ZAPOROJSTAL STEEL WORKS, *Russia* . . 66" Mill

THERE ARE MORE FARVAL SYSTEMS INSTALLED ON STEEL MILL EQUIPMENT THAN ALL OTHER MAKES COMBINED

THE FARVAL CORPORATION, 3278 EAST 80TH STREET, CLEVELAND, OHIO

Affiliate of The Cleveland Worm & Gear Company, Manufacturers of Automotive and Industrial Worm Gearing

IN CANADA: PEACOCK BROTHERS LIMITED

FARVAL

CENTRALIZED SYSTEMS OF LUBRICATION

CLEAN BILLETS.

they come fast from a BILLETEER

Chipping any heavy tonnage of billets by hand requires more than 10 times the man hours needed to chip the same tonnage with a Billeteer. The saving in cost is important - the saving in wasted man hours may soon be vitally urgent - speed is necessary.

With the Billeteer, bad heats can be reclaimed and normal production kept moving. As much or as little of the metal is removed as may be necessary and all steel removed is reclaimable. Seams and other defects are thoroughly removed - none are covered up.

Built for action, this machine wastes neither time, material nor man power. Every operation is power controlled, under the finger tip direction of a single operator.

Orders for Billeteers are filled according to priority - there are none in stock for immediate delivery, so we urge you to make your investigation and get your order in NOW.

BUILDERS OF BONNOT-LENTZ BILLETEERS ALSO CRUSHING PULVERIZING AND MIXING EQUIPMENT

THE Bonnot COMPANY
STEEL EQUIPMENT DIVISION
CANTON, OHIO

AIR... the Warrior !

Air—the magic carpet on which parachuters speed to battle! *Air*—the highway on which great 70 ton bombers roar to the substratosphere! *Air*—the transmission line through which radio's voice flashes! *Air*—that speeds production in countless plants vital to defense!

Yes, AIR is in the thick of the battle on many fronts—fighting, defense, production!

Imagine an entire 50 acre airplane engine plant—completely air conditioned! Well, there it is below—the world's largest. You can bet your boots it's sealed tight against dirt and prying eyes. And that's made possible by flooding the whole 50 acres with clean, comfortable conditioned air—circulated by 161 giant Sturtevant fans.

Then, there's the new Ford engine plant, where intricate engines that are marvels of precision flow from the assembly line like meat from a grinder. Here, to keep efficiency at a high pitch, Sturtevant equipment cleans and circulates 3380 tons of air per hour.

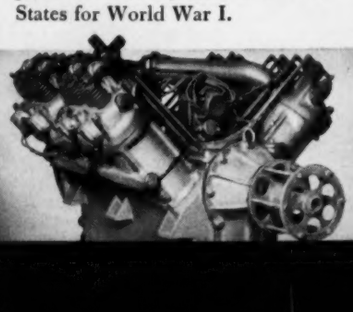
How about simulating actual flight conditions *on the ground*? Impossible? Not at all! See it in the famous Pratt & Whitney Engine test house—where Sturtevant air equipment is used for this purpose and to carry off lethal exhaust fumes.

These are merely typical examples. The list is endless.

Today—Sturtevant air handling equipment keeps airplane and engine plants humming—and ventilates and heats giant bombers. *25 years ago*—Sturtevant designed and built

Sturtevant
REG. U. S. PAT. OFF.
Puts Air to Work
in the Aviation Industry

"V-8" AIRCRAFT ENGINES made by Sturtevant as early as 1910, represented first use of aluminum alloys for airplane engine heads—and used on planes supplied to Britain, Russia and United States for World War I.



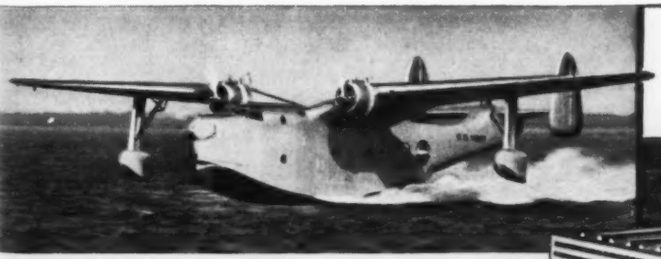
STURTEVANT "SCOUT" OF 1916—designed and built by Sturtevant Aeroplane Company, subsidiary of B. F. Sturtevant Company—the first plane incorporating steel construction and forerunner of today's all-metal planes.

the first planes to use steel construction. From the pioneer days up to the present time, Sturtevant has conspicuously served the Aviation Industry.

B. F. STURTEVANT COMPANY
Hyde Park, Boston, Mass. *Branches in 40 Cities*
B. F. Sturtevant Company of Canada, Ltd., Galt, Toronto, Montreal

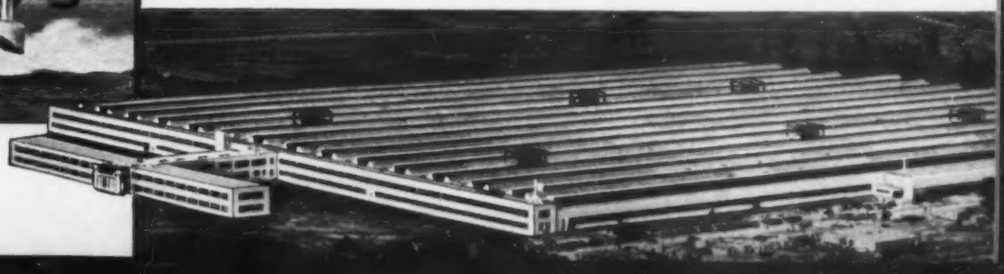
Sturtevant
REG. U. S. PAT. OFF
Puts Air to Work

STURTEVANT — FOUNDER OF THE AIR HANDLING INDUSTRY



ONE OF THE MODERN NAVY BOMBERS equipped with Sturtevant heaters and ventilators to maintain desired temperature and air conditions for officers and crew, regardless of outside weather.

50 ACRE, COMPLETELY AIR CONDITIONED Wright Engine plant at Cincinnati, Ohio, where 161 giant Sturtevant Fans circulate more than 7000 tons of air per hour!



WAGNER
HYDRAULIC BRIDGE BRAKES

are Standard Equipment on Cranes Manufactured by

THE CLEVELAND CRANE & ENGINEERING CO.

CLEVELAND CRANES

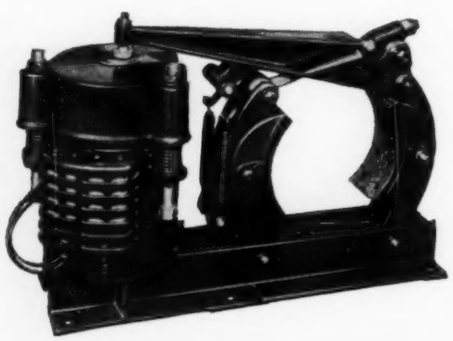
WICKLIFFE, OHIO

OTHER PRODUCTS
CLEVELAND TRAMRAIL
STEELWELD MACHINERY

Wagner Hydraulic Bridge Brakes are fast winning favor because of their sure, positive braking action, smooth stopping and controllable deceleration. They have a high safety factor and have a more positive braking action than mechanical brakes. Lost energy is eliminated.

Because of their rugged construction and thoroughly reliable actuation, The Cleveland Crane & Engineering Company has adopted Wagner Hydraulic Bridge Brakes as standard equipment on all of their cranes. The installation, illustrated at the upper right, shows how the compactness and streamlined appearance readily fits into the modern design of the Cleveland Cranes.

TYPE H Wagner hydraulically actuated bridge brake for inside cranes.



TYPE HM Wagner hydraulically actuated bridge brake with automatic parking attachment for outside cranes.

1 *This is the first of a series of advertisements designed to acquaint crane users with a few of the many crane manufacturers who are now standardizing on Wagner Hydraulic Bridge Brakes.*

WAGNER HYDRAULIC BRIDGE BRAKES HAVE MANY OUTSTANDING FEATURES

1...Easy one-point adjustment.

2...Exclusive anti-drag device.

3...They have hardened pins, knurled to prevent rotation.

4...They have bushings at points of maximum wear.

5...Grease fittings are provided on all hinge pins.

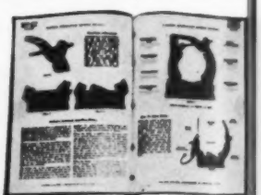
6...The friction blocks are extra thick and bolted to the shoes.

7...They are built by a manufacturer of Lockheed hydraulic brakes.

8...Wagner provides nation-wide service through twenty-five branches.

SEND FOR BULLETIN IU-20

This bulletin completely describes Wagner Hydraulic Bridge Brakes and should be in the hands of all production, safety and maintenance superintendents. Write for your copy today.



141-7

Wagner Electric Corporation

6400 Plymouth Avenue, Saint Louis, Mo., U.S.A.

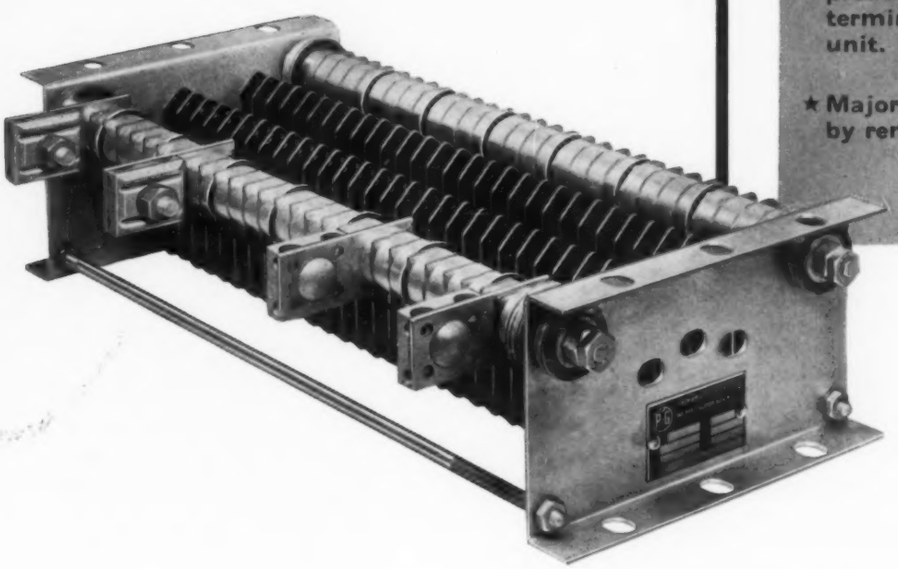
BRAKES • MOTORS • TRANSFORMERS • FANS

RESISTOR LAYOUTS *are Flexible*

with P-G... STANDARD RESISTOR UNITS

Little Effort Required To Make Adjustments

- ★ Minor variations often possible by wire change to other available terminals.
- ★ Greater variations easily produced by movement of terminal position in the unit.
- ★ Major variations effected by removal or addition of one or more standard banks.



OPERATING conditions today are changeable. To meet these changes it may be advisable to adjust the resistor layout to do the job better and quicker. P-G Standard Resistor Units already in use and a few spares in the supply room will help make these adjustments within a minimum time and with utmost efficiency. Bulletin No. 500, descriptive of these and other P-G Resistors is yours for the asking. A brief examination of this publication will explain why P-G is . . .

P-G

"The Resistor you can Install and Forget"

THE POST-GLOVER ELECTRIC COMPANY

• ESTABLISHED 1892 •

221 WEST THIRD STREET, CINCINNATI, OHIO

1-2-3-4-

SIX DIFFERENT PLANTS SELECT SC ONE-WAY FIRED SOAKING PITS IN SIX MONTHS..AN INVESTMENT OF OVER $1,500,000

● In the past six months six different plants selected or placed in operation Surface Combustion One-Way Fired Soaking Pits. Each plant made its own comprehensive investigation, visited all of the latest type of pits in operation and talked with the operators before a selection was made.

If you, too, are considering the installation of additional pit capacity, ask a Surface Combustion engineer to tell you about the latest improvements to One-Way Fired Pits. If you wish, arrangements can probably be made for you to visit one or several recent installations.

SURFACE COMBUSTION CORPORATION · TOLEDO, OHIO

"SECURITY FOR AMERICA"

SURFACE COMBUSTION

MANUFACTURERS OF INDUSTRIAL FURNACES · JANITROL GAS-FIRED SPACE HEATING EQUIPMENT · AND KATHABAR AIR CONDITIONING SYSTEMS

4-5-6

SYSTEMS

AIR RAID SHELTERS

PREPARE FOR SAFETY

Give your employees the maximum of protection with the new Wean bomb-proof air raid shelter.

Write today for complete information.

THE WEAN ENGINEERING CO. Inc.
WARREN, OHIO

PRODUCTS

- Continuous Strip Coiling Equipment.
- Special Conveying Mechanism.
- Automatic Pilers.
- Uncoiling Units.
- Press Feeding Equipment.
- Slitters.
- Strip Uncoiling and Automatic Shearing Units.
- Strip Coil Holders.
- Strip Coiling Reels.
- Sheet Galvanizing Equipment.
- Automatic Tinning Equipment.
- Automatic Doublers.
- Normalizing Furnaces.
- McKay Levellers and Processing Units.
- Strip Welding Equipment
- Continuous Strip Pickling Equipment including: Uncoilers, Up-Cut Shears, Stitching Machines, Pinch Roll Units, Recoilers, Drying Machines, Pickling Tanks, and Auxiliary Equipment.
- Scrubbing, Leveling, and Oiling Machines for Flat Products.
- Wilson Vertical Tube Type Annealing Furnaces.
- Corrugated Inner Covers.
- Automatic Feeding and Catching Tables.
- Continuous Pack and Pair Heating Furnaces.
- Wheelabrator Abrasive Cleaning Equipment.
- Automatic Weight Classifier.
- Automatic Gauge Classifier.

ONLY
Cables Insulated with
OKONITE
Give You ALL 5

1 Long-lived insulation, made with the world's finest rubber —wild up-river fine Para.

2 Conductors better protected from corrosion by an Okoloy coating over the copper.

3 Foot-by-foot inspection of the insulation strips before and during their application to the wire.

4 Perfect centering of conductors by the Okonite strip-insulating process.

5 Greater density, uniformity and higher electrical values through vulcanization in a continuous metal mold.

OKONITE insulated wires and cables alone provide these 5 important features in all sizes, types and constructions. Our Engineering Service Department is at your service for consultation on any wire or cable problem.

THE OKONITE COMPANY
PASSAIC, NEW JERSEY
Offices in principal cities

OKONITE

Specify OKONITE
INSULATED WIRES & CABLES

FATE DICTATES -- AETNA ANTICIPATES

There are few today that can forecast the future with any certainty. We do know that stupendous industrial production is the present requirement. For the past two years we have been busy enlarging our plant capacity, re-tooling and increasing our personnel. Thus we demonstrate our faith in America's Future. . .

Main Bay
Ellwood City Plant

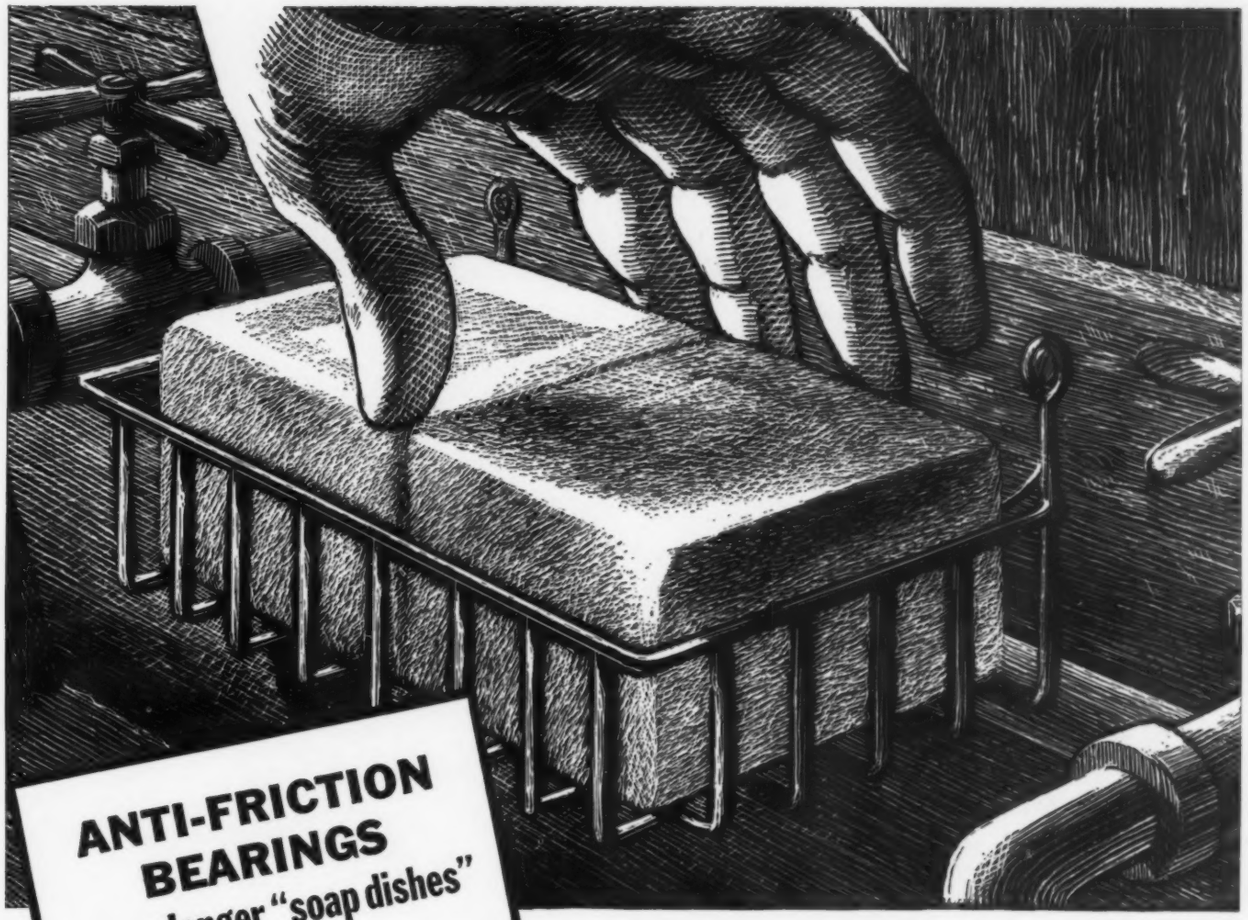
Main Bay
Warren Plant

The **AETNA-STANDARD**
ENGINEERING COMPANY
★ YOUNGSTOWN, OHIO, U.S.A. ★

ASSOCIATED COMPANIES

HEAD, WRIGHTSON & COMPANY, LIMITED, THORNABY-ON-TEES, ENGLAND • AKTIEBOLAGET ARBOGA MEKANISKA VERKSTAD, ARBOGA, SWEDEN • JOHN INGLIS COMPANY, LIMITED, TORONTO, ONTARIO, CANADA

Designers and Builders to the Steel, Non-ferrous and Chemical Industries



. . . after changing to TYCOL GREASES

ANTI-FRICTION bearings were so caked with soap deposits from easily broken down grease that they could pass for "soap dishes." This accumulation of old, hard, soapy grease prevented the entrance of new lubricant, caused costly damage to the roller bearings from lack of proper lubrication. But the bearing "soap dish" era has gone and bearing worries are ended now that this steel mill is using Tycol Grease.

Why do Tycol Greases stand up in tough lubricating jobs where other greases soon run out or separate to form hard soap deposits? It's because Tycol Greases are made from the highest quality lubricating oils combined with a minimum of soap for a given consistency. Tycol Grease is water repellent, "stays put" indoors and out.

A Tide Water engineer will gladly help you maintain maximum production through proper lubrication. Full details may be obtained by writing the Tide Water Associated Oil Company, 17 Battery Place, New York, N.Y.

DRUMS, DRUMS, DRUMS

Defense needs make it extremely important that all empty drums be returned immediately.

Regional Offices: Boston, Philadelphia, Pittsburgh, Charlotte, N.C.

TIDE WATER ASSOCIATED OIL COMPANY

Eastern Division: 17 Battery Place, New York, N. Y.

INDUSTRIAL TYCOL OILS-GREASES

TIDE WATER INDUSTRIAL LUBRICANTS

THERE IS A COMPLETE LINE OF TYCOL LUBRICANTS SCIENTIFICALLY ENGINEERED FOR EVERY INDUSTRIAL USE

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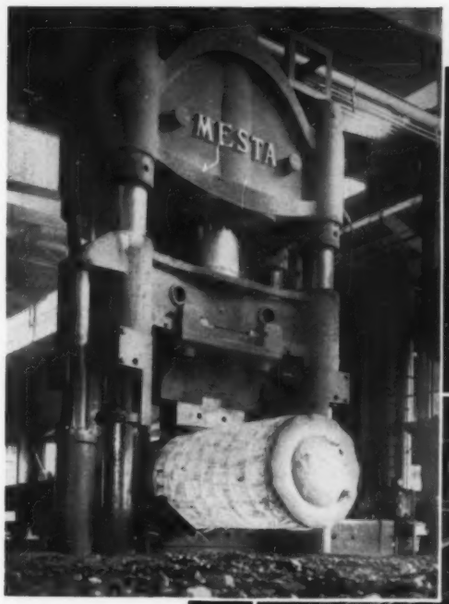
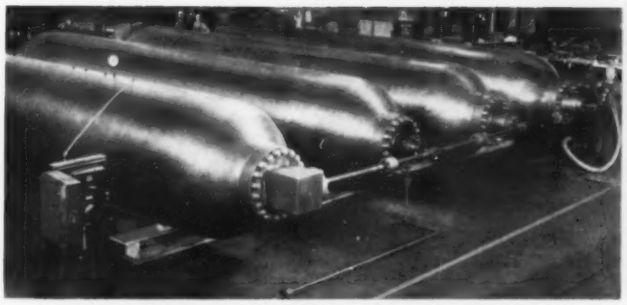


MESTA FORGINGS

Forging 100" Ingot in 6,000 Ton MESTA Forging Press

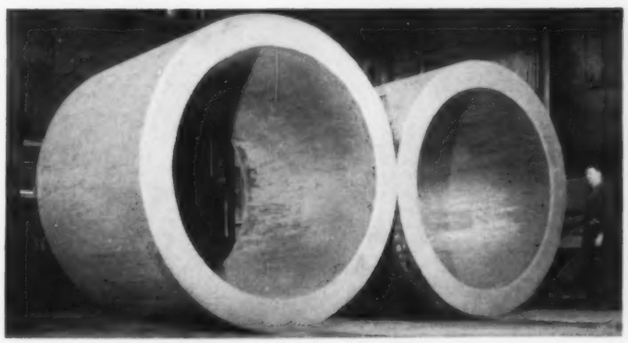
View in MESTA Ship Shaft Department

MESTA was among the first to be awarded the FLAG of the UNITED STATES BUREAU OF ORDNANCE and NAVY "E" PENNANT.



MESTA Forged Steel High Pressure Vessels

View in One of MESTA Small Lathe Shops



Forged Ring Gears for Marine Turbines

Forged Steel Column for 14,000 Ton Armor Plate Press

MESTA MACHINE COMPANY · PITTSBURGH, PA.

"Hay!" We are not this slow!

Our expanded facilities and efficient organization offer you . . . . . .

Remarkably fast deliveries on North American Burners, Mixers, Regulators, Valves and Accessories.

If you are looking for dependable Combustion Equipment in a hurry, we invite you to try

Combustion Equipment by

NORTH AMERICAN

THE NORTH AMERICAN MANUFACTURING COMPANY

MANUFACTURERS OF INDUSTRIAL FUEL BURNING EQUIPMENT FOR GAS OR OIL

CLEVELAND, OHIO

BRANCH OFFICES WITH FACTORY TRAINED REPRESENTATIVES IN PRINCIPAL CITIES

MANHATTAN RUBBER helps the flow of *Steel* into a thousand shapes—for Defense ★ ★ ★

The sparks and the glow and the lurid liquids that flow from furnace and ladle are the climax of grandeur in the saga of steel.

But this molten spectacle is but one scene in the solid drama of metallurgy where rubber plays a varied role.

MANHATTAN makes hundreds of products which help shape billet and bloom to the needs of Man as well as of Mars.

Ore and coal, coke and limestone travel from mine and quarry on rubber conveyor belts; air and water and steam are borne through rubber hose; parts are welded, sand-blasted, pickled and plated with the aid of rubber. These and an astonishing list of other products—like abrasive wheels for snagging and cutting off, and asbestos brake lining for cranes and hoists—are part of what MANHATTAN makes for the Steel Industry under the banner of Defense.

★ ★

A few MANHATTAN Products used by the Steel Industry

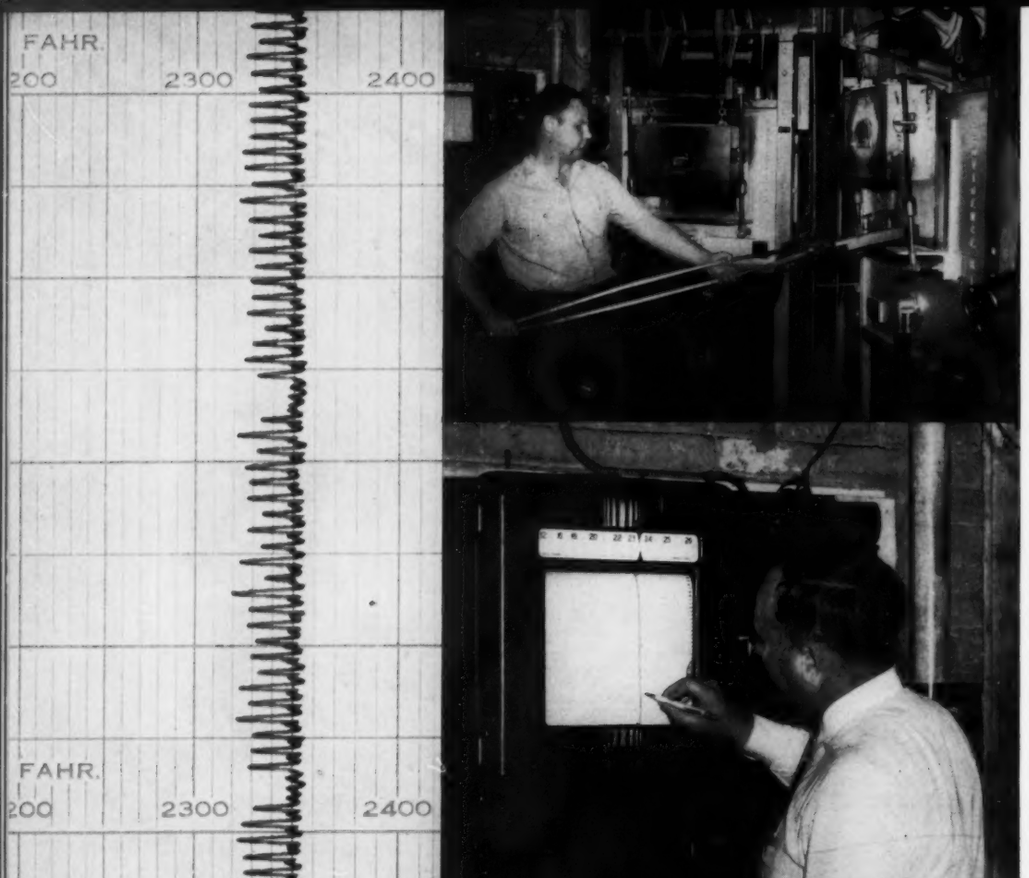
Conveyor Belts, Mildew-proof
Rubber Belting
V-Belts
Hose for Acid, Air, Chemicals, Sand-Blasting, Steam, Water and Welding
Fire Hose
Furnace Door Hose
Rubber-Lined Pickling and Plating Tanks
Rubber Covered Rolls
Abrasive Wheels
Brake Lining, Clutch Facings
Asbestos Cable Covering
Asbestos Products
Molded Goods and Packing
Rubber Pipe

★

● A typical Manhattan conveyor belt installation in the steel and allied industries.

KEEP AHEAD WITH MANHATTAN

THE MANHATTAN RUBBER MFG. DIVISION
of RAYBESTOS-MANHATTAN, INC.
GENERAL OFFICES and FACTORIES—PASSAIC, NEW JERSEY



Each "spike" on the Micromax Pyrometer chart at left means that one more batch of milling cutters has gone into the Micromax-controlled heat-treating furnace. Charging is shown in photo at top. Above, the foreman glances at the Pyrometer to see that all is going well.

HEAT-TREATING IS IMPROVED

Because Micromax Pyrometer Can "Jump"

Sensitive and accurate control of temperature is of course not new to the Morse Twist Drill & Machine Co., but it is more easily secured now than ever before, in the heat-treatment of high-speed steels, because the Company is using a Micromax Pyrometer, with a Rayotube as temperature detector.

Milling cutters and similar tools are preheated before they enter the Company's Hayes high-speed-steel furnace, which the Micromax Pyrometer is holding constant at the high hardening temperature. The batch, plus the holder, weighs only a few pounds. Nevertheless, when it enters the furnace, the Micromax pyrometer is so sensitive that its record actually moves off at a right angle; it seems almost to jump. And the instrument simultaneously calls for more heat energy. As temperature rises in response to this call, the Micromax watches carefully, and shuts down on the supply in such a way as to return temperature to the control point with promptness and precision. So successful is the control that rejects due to heating are almost unknown.

The Micromax sensitivity and responsiveness, which perform somewhat spectacularly here, are available for any type, size or make of furnace, or almost any other heat-using equipment. If you have a temperature-control problem, outline it and we will send the appropriate catalog.

After hardening, this die is used by the Gray Manufacturing Co. to stamp their company name on telephone coin boxes.

Finely-Engraved Dies Deserve Vapocarb-Hump Hardening

The question of how to handle this die in the hardening furnace might well be a problem in many heat-treats. Its original dollar-a-pound cost, for the steel, was of course redoubled several times as highly-skilled die-sinkers labored over it, so that a bit of uneasiness as to furnace operation might well have been expected.

But that wasn't the case with this tool. The Gray Manufacturing Company, makers of telephone coin-boxes, use the Vapocarb-Hump hardening method on all tools like this, and there's no reason for uncertainty in such cases.

For the heat-treater has, in the Vapocarb-Hump Method, everything he needs to assure successful handling of the die in the furnace. He can shut the air away entirely, by means of Vapocarb atmosphere, thus eliminating all possibility of scale, pits, soft spots, and decarburization. He can see the temperature of the

Heat-treater loading die into Vapocarb-Hump furnace.

die, the temperature of the furnace, and the difference between them, and can control this latter to prevent warp. And the die itself will tell him when it enters the critical, and when it leaves, so that he can decide just when to quench it to secure the desired depth of hardness. Undaunted by the high value of the tool, the heat-treater can thus proceed to harden it so as to give it the longest possible life—many years, in this case.

If your plant suffers from erratic performance of tools — even of low-priced ones — Vapocarb-Hump hardening can probably help you. Ask for Catalog T-621.

Jrl Ad ENT-0600C(34)

LEEDS & NORTHRUP COMPANY, 4942 STENTON AVE., PHILA., PA.

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MEASURING INSTRUMENTS · TELEMETERS · AUTOMATIC CONTROLS · HEAT-TREATING FURNACES